# POWER

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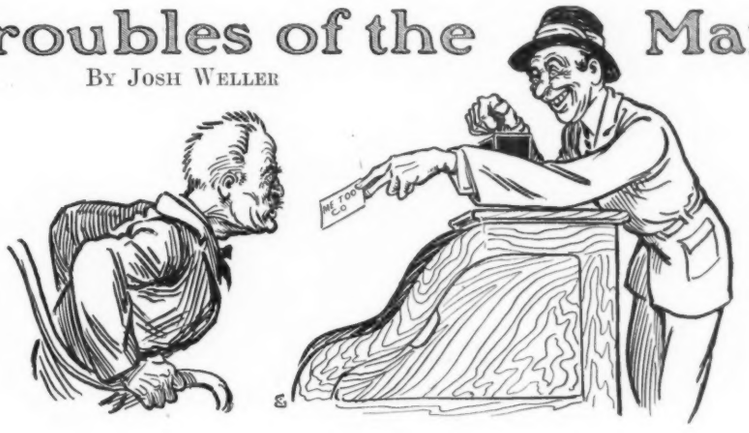
## Troubles of the Manager

By Josh Weller



THE salesmen "lay" for me at night, they wait for me at morn;
I know they will be on my trail when Gabriel blows his horn.
They take my time when I should work, they do not seem to care;
I have no chance to change my pants, no time to pray or swear.
Oh, Holy Moses, Holy Smoke and Holy Mack'rel, too,
Look down at once and tell me quick, what can I, should I, do?

I BOUGHT a monster bulldog once, with teeth like Teddy R.;
I tied him at my office door and left that door ajar.
I chuckled low, I chuckled long, and then I chuckled some;
I said, "We'll see some royal sport when those fresh salesmen come."
They came at last, they came in force, that bloody salesmen gang;
They had a dentist with them, and he pulled each blooming fang.

I BOUGHT a pound of strychnine once and put it on some meat,
The rascals sniffed and smelled of it; they were too wise to eat.
I nearly killed a salesman once, a sassy red-haired runt,
He talked of testing furnace gas or some such crazy stunt.
I broke his neck, I broke his back, I broke his head and slats;
I threw him on a garbage pile to feed the dogs and rats.

NEXT morning when I came down town my eyes popped from my head,
The salesman on the garbage pile had risen from the dead.
He stayed and talked and talked and stayed and used up all my day;
At last I bought his worthless junk, there was no other way.
I sat before my fire one night to warm my frosted foot,
A salesman cuss came down the flue and filled the house with soot.

HE talked of scale and soot and ash and said my tubes were bad,
My heating surface punk, or worse. He sold me all he had.
Then when I got my nightie on, my "Now I lay me" said,
I found another salesman hiding underneath my bed.
There is no joy in life for me, no rest where'er I go,
Some salesman's always butting in, this world's a vale of woe.

AND when at last I shuffle off and hike for Peter's gate,
I'll find some salesman waiting for me there as sure as fate.
And if I give the scamp the slip and get inside the wall,
He'll steal St. Peter's golden keys and catch me after all.
Ha, ha, ha, ha, ho, ho, ho, ho, and likewise hee, hee, hee,
At last I know what I shall do, just watch my smoke and see.

I'LL buy a ticket right straight through to Satan's warm domain;
The bunch will all be after me, they'll all be on my train.
I'll bribe the Devil and his imps, I'll bribe them till I'm broke,
But I'll get those salesmen in the pit and pile on lots of coke.
And when one tries to scramble out he'll find me standing by,
Armed with a red-hot pitchfork, and I'll jab him in the eye.

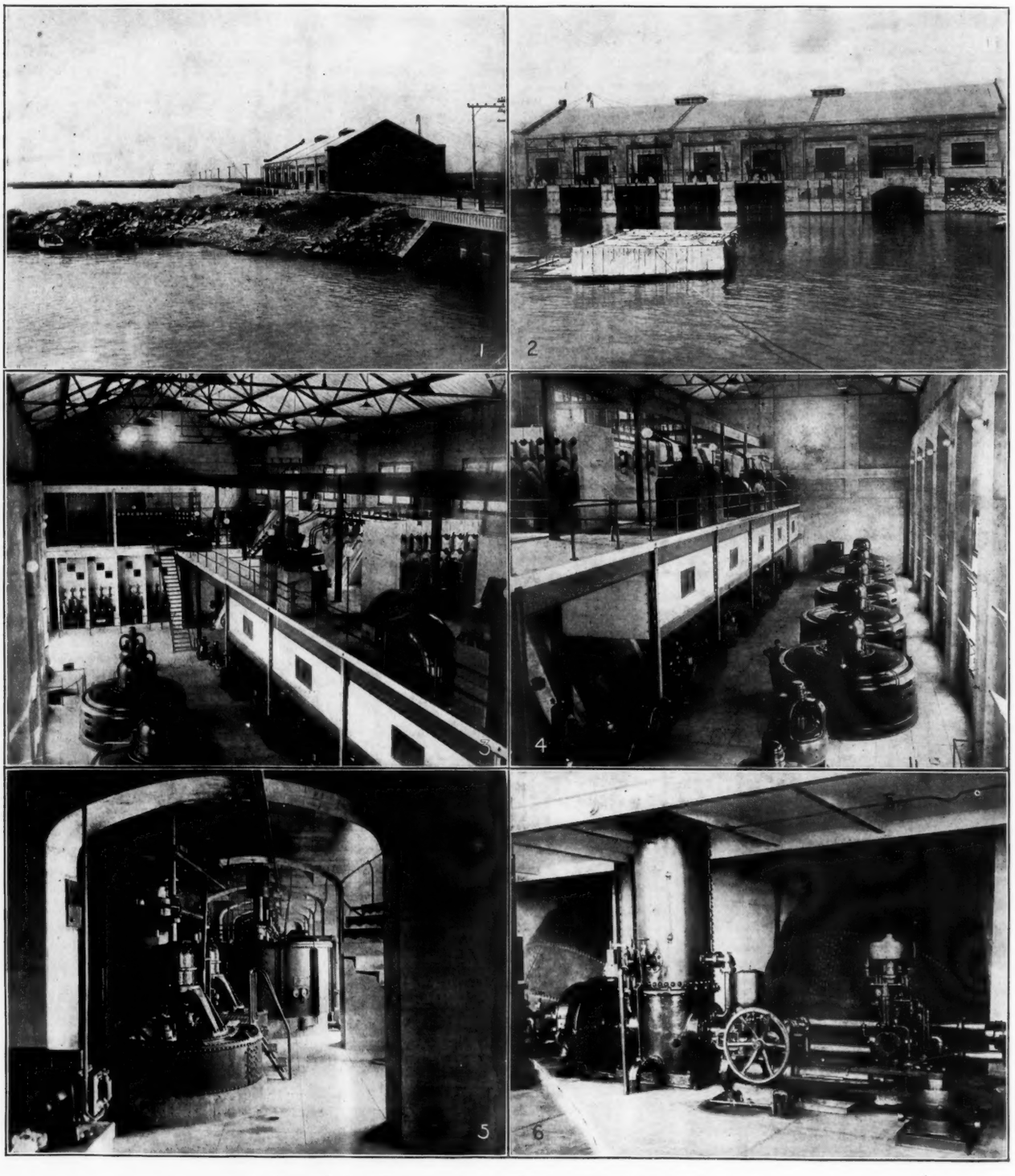
*Read the Above to the Next Salesman That Calls on You*

# Federal Project at Minidoka, Idaho

By A. P. Connor

*SYNOPSIS—The Minidoka project of irrigation consists of the Snake River dam to impound the water which supplies two main canals and which is also used to operate five 1200-kw. turbo-generator units at the power house. The current generated is transmitted to four pumping stations at distances of from twelve to twenty miles from the power plant, elevating the water from the main canal into canals at a higher elevation. The pumps have a capacity of about 125 sec.-ft. each.*

The Minidoka project, situated in the counties of Lincoln and Cassica, Idaho, is typical of some of the Federal



Figs. 1-6. Views of the Minidoka Power Plant

Government's undertakings requiring power for operation. In this case a storage dam 4300 ft. long impounds the water of an 18,000-square mile watershed and is supplemented by diversion dams 600 ft. long. The dams are of the earth- and rock-fill type, having an average height of 50 to 52 ft., and raise the water level sufficiently to supply two systems of canals having an aggregate length of 130 miles, and supplying 190 miles of smaller canals.

In addition, power is taken from the impounded waters hydro-electrically and transmitted to pumping stations at the terminals of these main canals where the water is pumped to higher levels for distribution to branch canals. At present about 6000 kw. is generated. This power is to supply water to 70 miles of main and 60 miles of branch canals by means of pumping stations located at distances of from 12 to 20 miles from the dam. The water is stored in Lakes Jackson and Walcott, and the average run-off is about 7,200,000 acre-feet. The altitude of the project is 4200 ft. above sea level. The hydro-electric generating units work under an average head of 35 ft.

In Fig. 1 is shown the main-dam, power house and the high-tension transmission line, looking from Lake Walcott. The power house is built of concrete and is a part of the dam; its general appearance from the intake side is shown in Fig. 2. Interior views are shown in Figs. 3, 4, 5 and 6. The maximum inside width of the building is 45 ft., and the usable width for the generators is about 20 ft., requiring a somewhat contracted arrangement of the units and apparatus.

The sub-basement of the power house is in reality a tunnel space for the passage of water through the dam whenever the lower gates of the dam are opened. The basement provides for the hydraulic portion of the plant, such as the penstocks, turbines and incidental casings. On the main floor are the generators, auxiliary apparatus and accessories. The gallery is for the switchboard, transformers, high-tension switches and the like.

The main transformers are in the power station and step up the voltage of the generators from 2300 to 33,000 volts, the connections being delta on the low-tension side and Y or star on the high-tension side. A spare transformer is provided for emergencies. The transformers

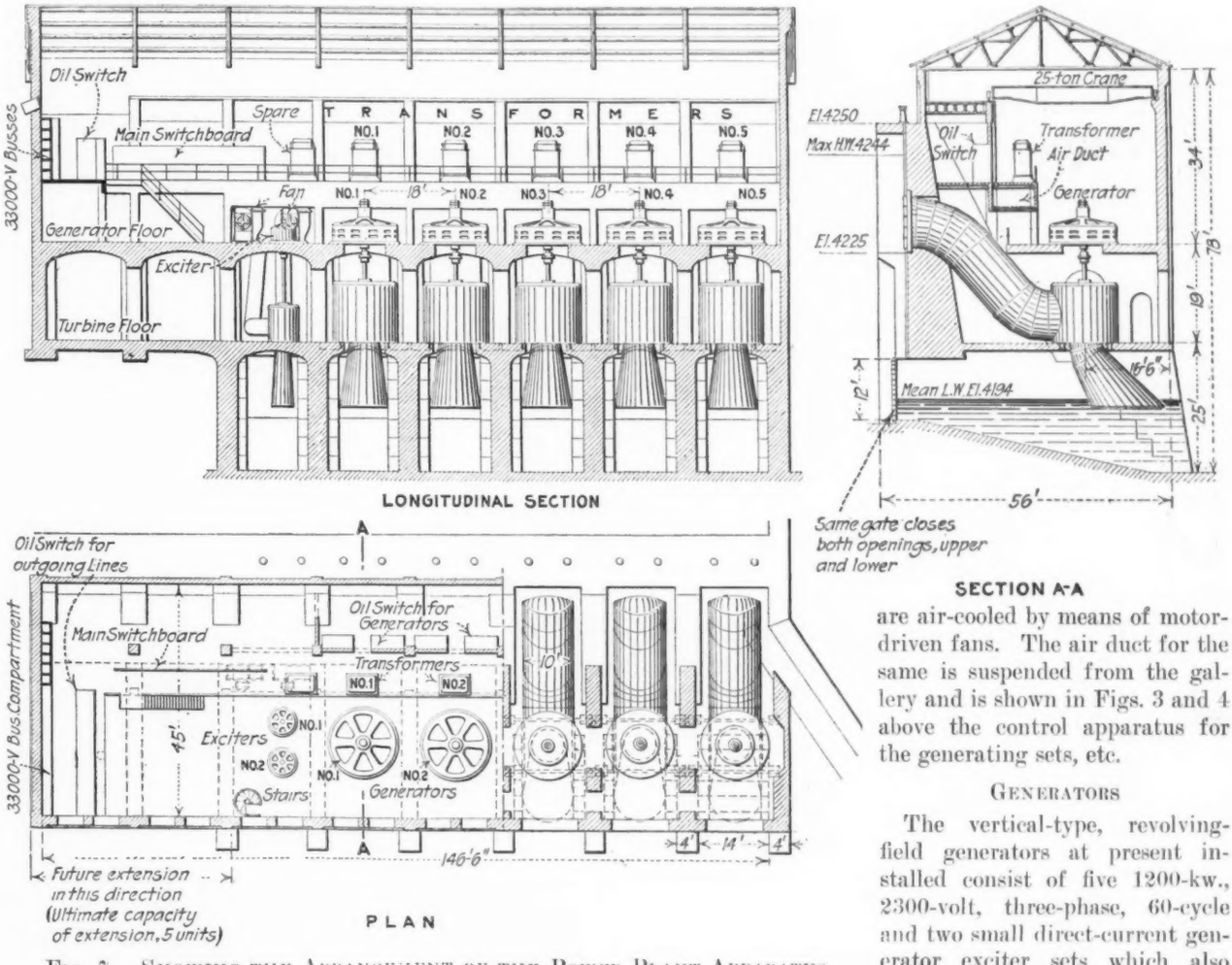



FIG. 7. SHOWING THE ARRANGEMENT OF THE POWER-PLANT APPARATUS

are air-cooled by means of motor-driven fans. The air duct for the same is suspended from the gallery and is shown in Figs. 3 and 4 above the control apparatus for the generating sets, etc.

### GENERATORS

The vertical-type, revolving-field generators at present installed consist of five 1200-kw., 2300-volt, three-phase, 60-cycle and two small direct-current generator exciter sets which also run the station motors. The rated capacity of each alternator is 1400 kv.-a. at 85 per cent. power factor at a speed of 400 r.p.m. The generators are arranged on the main floor and connected with their turbines by extended shafts. The 120-kw. exciters are compound-wound, for 125 volts at 425 r.p.m.

### TURBINES

The 1800-hp. main turbines are of the inward-flow, axial-discharge, single-runner type fitted with pivot gates. Those for the exciters are 180-hp. All of the turbines are between the arched columns in the basement, and ladders and passageways enable every part to be reached easily for inspection or repair.

The turbines are arranged for low head and are supplied with penstocks 10 ft. in diameter directed toward the turbines at a 45-deg. angle, with curved ends to reduce friction. Fig. 6 is a view of the governing apparatus. The penstocks are practically self-supported and are uncovered. The larger ones are closed by gates weighing 5 tons each, and operated by 6-hp. motors which receive energy from the 125-volt, direct-current busses. The gates for the exciter turbines are operated by a 2-hp. motor. The mechanism of the gates is outside the power station over the gates and is inclosed. The gates are arranged to close the upper and lower outlets in the dam so that when the upper one connecting with the penstock is open the lower one opening through the dam will be closed, and when the latter is open the former will be closed. This enables each gate to do double duty and avoids the need of two sets. They may be operated by hand when desired.

The draft tube of each turbine is enlarged at the end to improve the discharge. The penstocks and draft tubes are made of boiler iron in riveted sections. The station is provided with a 25-ton traveling crane for handling heavy pieces.

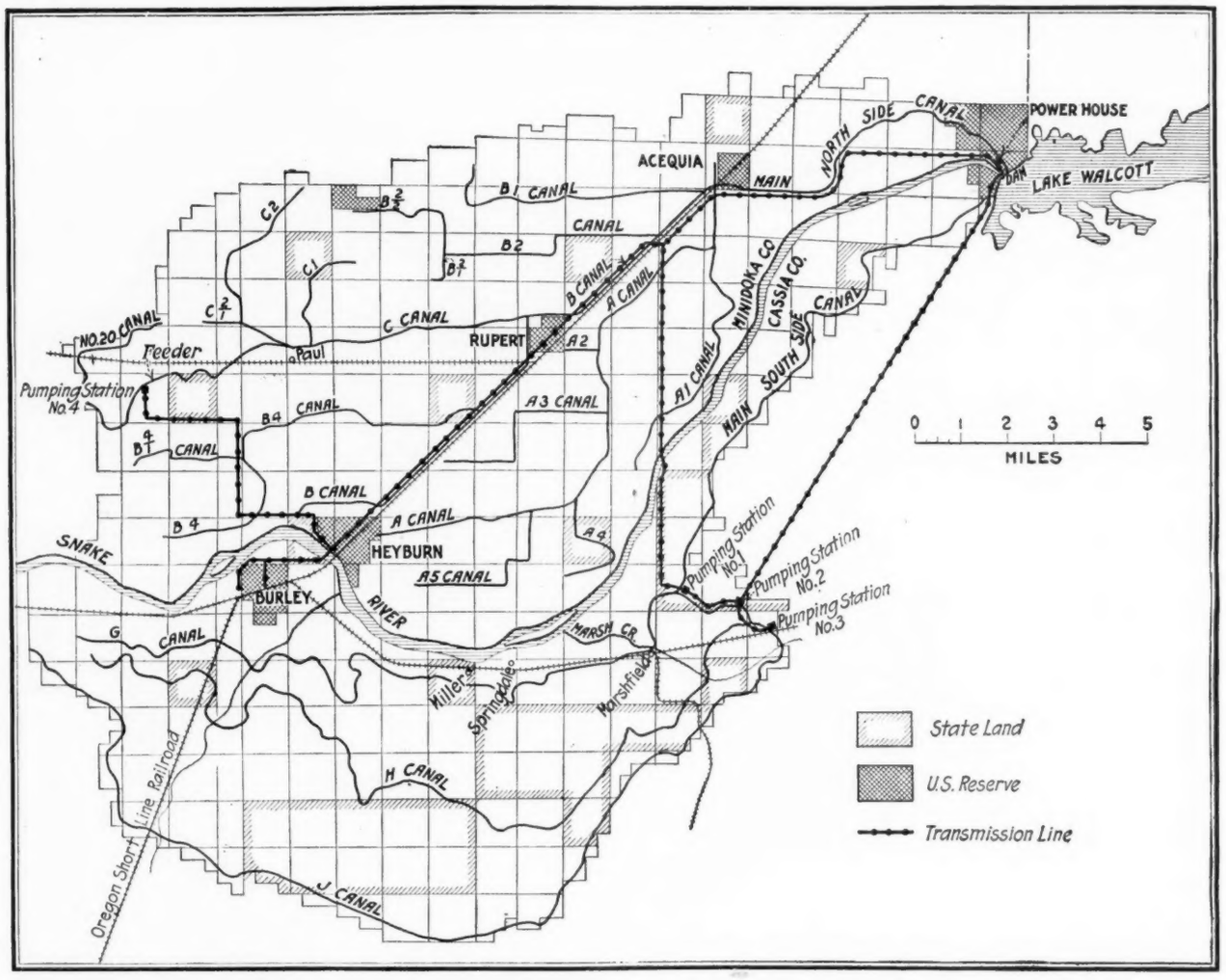



FIG. 8. MAP OF THE MINIDOKA PROJECT, SHOWING THE AREA, ITS CANALS AND TRANSMISSION LINES

## GENERAL

There are two main canals running from the dam, known as the Main North-Side Canal and the Main South-Side Canal, and these are filled with water by gravity. The latter is about 15 miles long and terminates adjacent to canals known as G, H and J, which are situated on higher levels. A transmission line is run 12 miles across the country from the power station to these canals to feed the motors at the pump houses. The pumping station No. 1 takes the water from the Main South-Side Canal and raises it to the level of Canal G, which is about 20 miles long. About a mile distant water is taken from Canal G and raised to Canal H through No. 2 pumping station. Canal H is about 25 miles long, and Canal J is fed with water from H through another station, No. 3. Pumping station No. 4 is on a transmission line on the north side of Snake River and takes water from the North-Side Canal. This station is about 20 miles from the dam, by air line, and about 30 miles by following the transmission lines. These are shown in Fig. 8. The layout of the system is such that the transmission lines are tied in and are run to include the various towns on the project. The main object of the power is, however, the pumping.

The pumping stations are buildings of substantial dimensions and are equipped with ample pumping units for the duty expected at each location. Each station has air-cooled step-down transformers. Exciter and motor-generator sets are provided for the minor direct current required for small motors and lighting in the pumping stations.

The ultimate capacity of the pumping stations in pumping units is as follows: No. 1, four pumping units, each of 125 sec.-ft. capacity; No. 2, three pumping units, each of 124 sec.-ft. capacity; No. 3, one pumping unit of 125 sec.-ft. capacity; No. 4 is about the size of that for No. 3.

Details of a pumping unit are shown in Fig. 9. It is driven by a 2200-volt synchronous, three-phase, vertical motor. The pumps are of the centrifugal type, taking the water centrally and forcing it out peripherally. The pumps are arranged to be always submerged, and the amount of water supplied and raised by each is controlled by gates with lips, which close the ingress openings to the pumps. The gates in each case are operated through a lever mechanism actuated by a float disposed in the incoming water, Fig. 9.

A baffle plate is provided above the entrance of each pump to keep out obstructions and to serve as a guide for the lip-operating rods. A grating in front of the water

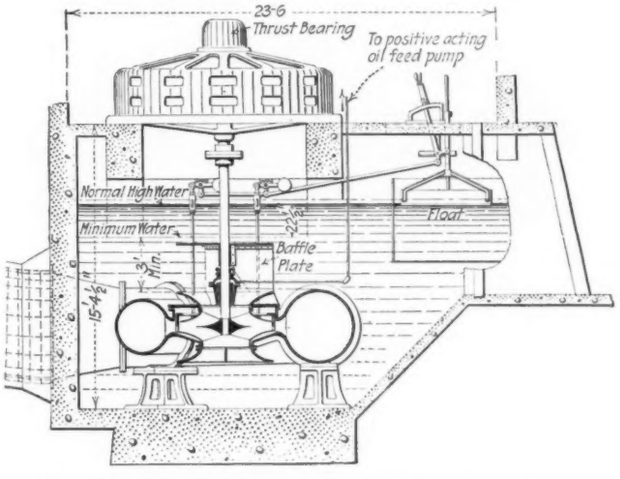



FIG. 9. VERTICAL SECTION OF A PUMPING UNIT

entrance of each pumping station keeps out general rubbish and ice. The float mechanism has a lever and rack for raising it independently of the water.

The water raised by the pumps is diverted through an upwardly curved pipe or duct, from which it goes to its respective canal.

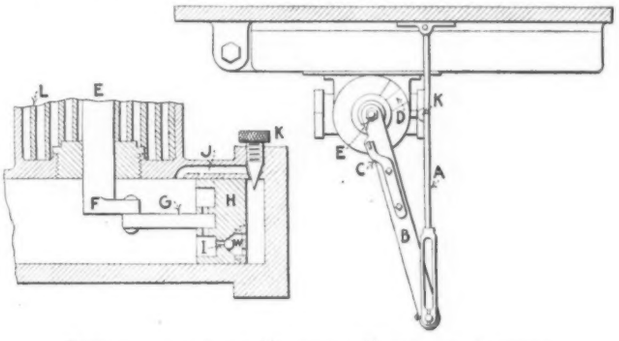
## Morrow Furnace System

The Morrow system is an automatic furnace door-closing device to assist combustion, to obtain the maximum amount of heat from the fuel and to eliminate smoke. The equipment consists of a check with a supplementary device for holding the door of a furnace wide open while the fireman is putting in the fuel, and which on the release of the catch allows the door to swing to a closed position, hesitating long enough when partly closed to allow an inrush of the required amount of air necessary for combustion.

The checks are mounted upon the furnace door and are operatively connected by the links *A* (see illustration), with the boiler front. *B* is the operating arm, which is connected to the link *A* at one end and to the retarding mechanism at the other. *C* is a spring detent, which, when the furnace door is open, will engage within the notch *D*, formed in the upper surface of the casing of the device, so as to retain the door in an open position. In the sectional view through the closing device, the vertical shaft *E*, operated by the arm *B*, is provided with a crank *F* at its lower end, which is connected to the link *G* with the piston *H*. A check valve *I* opens to admit the retarding fluid when the furnace door is opened, but prevents the fluid from passing through the piston when the piston moves in a direction to close the door.

The bypass *J* extending around the piston is controlled by the needle valve *K*, by the adjustment of which the flow of liquid is governed through the bypass, which regulates the speed at which the door will close.

When the furnace door is opened it moves the upper end of the casing of the closing device relatively to the spring detent *C*, until the latter engages within the notch *D*, thereby holding the door open. After firing, the door is given a slight push inwardly, which unseats the detent




DETAILS OF THE MORROW FURNACE SYSTEM

*C* and permits the spring *L* to close the door. The operation of the spring is controlled by the movement of the piston *H*, which is governed by the bypassing of the fluid through the passage *J*.

This device is the invention of J. H. Morrow, chief engineer, Great Northern Hotel, Chicago, Ill.

## Rules for Weight of Cast-Iron Pipes

The usual method adopted for calculating the weight of cast-iron pipes consists in finding the cubic contents of the metal in inches, and multiplying that by the weight of one cubic inch of cast iron = 0.26 lb.

EXAMPLE—Taking a pipe 12 in. internal diameter, ½ in. thick, 9 ft. long, we have outside diameter 13 in. = 132.732 sq.in. area, internal diameter 12 in. = 113.097 in. area.

132.732 — 113.097 = 19.635 sq.in. sectional area.
9 ft. = 108 in., hence
19.635 × 108 = 2120.58 cu.in. of cast iron, and
2120.58 × 0.26 = 551.35 lb.

The two flanges, or one socket, are usually reckoned equal to one foot length of pipe.

Another formula for calculating the weight of cast-iron pipes is

$$W = 2.45\ (D^2 - d^2);\ \text{or}\ 2.45\ (D + d) \times (D - d)$$

in which

W = Weight per lineal foot in pounds;
D = External diameter of pipe in inches;
d = Internal diameter of pipe in inches;
2.34 = A constant.

A very useful approximate rule for the weight of cast-iron pipes is, for a pipe 9 ft. long, with flanges at each end, and 1 in. thick, allow 1 cwt. for every inch in diameter, keeping the thickness and weight proportional, either more or less.

EXAMPLES—A 12-in. pipe, 9 ft. long, flanges at each end, 1 in. thick, will weigh approximately 12 cwt.

A 12-in. pipe, 9 ft. long, flanges at each end, ¾ in. thick = 9 cwt.

A 10-in. pipe, 9 ft. long, flanges at each end, ½ in. thick = 5 cwt.

—Björling.

# Small Condensing Turbines

By W. J. A. London*

*SYNOPSIS—An unusually interesting article on the considerations entering into the design of small condensing turbines, with particular reference to the Terry "return-flow" machine.*

Since the introduction of the small direct-connected turbine on a commercial basis, some eight years ago, until recently, fully 90 per cent. of the machines called for were intended for noncondensing service. In the few cases where they were called upon to operate condensing, such as for marine work, little attempt was made at economy, as the operation of these machines condensing was more a matter of convenience than of water rate. It has been acknowledged that the designing of small turbines is much different from that of large machines, for were a small turbine designed on the same principles and lines as a big machine, a hopeless commercial failure would result. There have, therefore, been two distinct fields in turbine work; the principles governing the designs of small and large machines being so much at variance that they might be said to be almost as different as the designs of a steam and a gas engine.

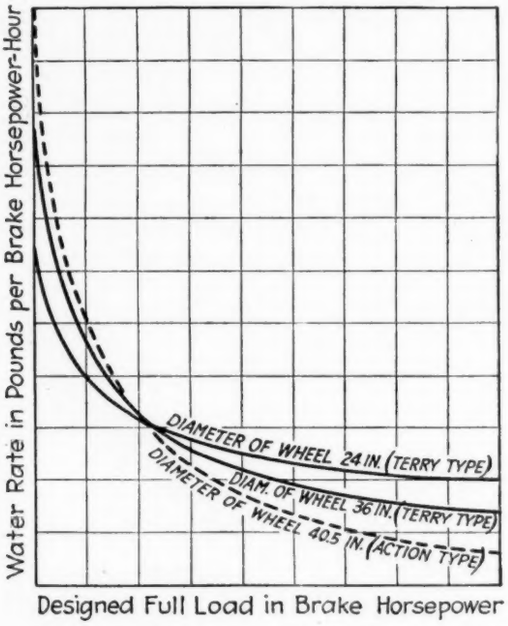

Water Rate in Pounds per Brake Horsepower-Hour
DIAMETER OF WHEEL 24 IN. (TERRY TYPE)
DIAM. OF WHEEL 36 IN. (TERRY TYPE)
DIAMETER OF WHEEL 40.5 IN. (ACTION TYPE)
Designed Full Load in Brake Horsepower

FIG. 1. RELATIVE EFFECTS ON WATER RATE OF VARYING DIAMETER OF WHEEL AND DESIGNED FULL LOAD

The characteristics shown are due to increased windage losses on the larger wheels. All curves are plotted for the same conditions of steam pressure.

Most small turbines were installed to operate noncondensing, being used primarily for auxiliary apparatus, the exhaust being used in feed-water heaters. Small isolated plants were operated noncondensing, the exhaust steam being used for industrial or heating purposes.

For isolated plants, such as small pumping installations of, say 150 to 500 hp., where economy was of much importance and the saving by operating condensing sufficient to offset the first cost, maintenance, etc., the turbine has been at a disadvantage as compared with the reciprocating engine.

The average thermal efficiency of small turbines is in the neighborhood of 40 per cent. With this efficiency all exhaust steam can be utilized without "blowing off," so that no higher efficiency is required or even desirable.

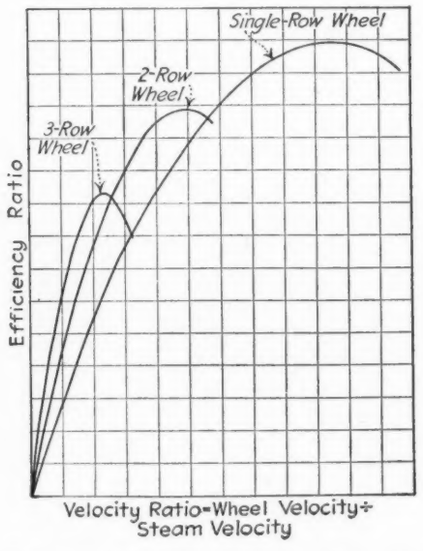



FIG. 2. RELATIVE EFFICIENCIES OF IMPULSE WHEELS WITH VARIOUS NUMBERS OF ROWS OF BUCKETS

Forty per cent. efficiency does not represent the highest available in this class of machine, but it is conceded to be about the highest commercial efficiency. The cost of small turbines varies approximately directly as the square of the diameter of the runner, whereas the efficiency is increased approximately only inversely as the square root of the runner diameter, so any saving in steam consumption must be accompanied by a marked increase in first cost.

Above 500 to 600 hp. the field of the large turbine, where water-rate efficiency is of paramount importance, is approached. These machines operate condensing in the same proportion that the small machines operate noncondensing, and the whole problem of design must be attacked on a totally different fundamental basis.

For powers of, say 150 to 500 hp., condensing, where high efficiencies were desirable, the reciprocating engine until recently had no serious competitor in the steam turbine. The reason for this is obvious. The reciprocating engine was developed, and the meager demand for small high-efficiency turbines did not warrant the manufacture of special machines, and furthermore, the customer would not pay the price that would have to be charged.

Conditions have changed rapidly during the last two years or so, and there is now a big demand for small condensing turbines of high efficiency, both high- and low-pressure, which has led to the development of a third class of machine to meet the requirements of this market. To distinguish this class from the small and the large machines, it is permissible to call it the "in-

*Chief engineer, the Terry Steam Turbine Co.

termediate design." This design should have, as far as possible, the simplicity and accessibility of the small machine with an efficiency approaching that obtainable with the larger units.

When Parsons and DeLaval built their first turbines the main trouble was not with the turbine itself, but with the "other end," or the driven unit. A turbine is of little use by itself, and it was not until it was demonstrated that it had come to stay that generator, pump and blower makers awoke to the fact that they must remodel their apparatus to meet turbine requirements.

Rapid as the turbine development has been, it would have been more so had it not been for the slow development of the "other end." And past events have again repeated themselves in the field of the "intermediate design." This machine would not have been possible had it not been for the rapid strides that have taken place along the following lines: (a) The manufacture at reasonable cost of high-tensile steel for turbine wheels; (b) the increase in permissible speed of generators, blowers, etc.; and (c) the introduction on a commercial basis of the speed-reducing gear.

In POWER of Oct. 28, 1913, a brief description was given of the return-flow condensing steam turbine that had just been developed by the Terry Steam Turbine Co. for this so called "intermediate field."

On the completion of tests of the first machine, several modifications and improvements naturally suggested themselves and are incorporated in the latest designs. For turbines of small power the latest tests show some remarkably good efficiencies, as will be seen by the details of the tests published herewith.

One of the main changes in design has been to carry the principle of the horizontally divided case to the last extreme. In the original return-flow machine the casing was divided horizontally, but the center diaphragms and the center-diaphragm glands were not, whereas in the machine shown all diaphragms, and diaphragm and end glands are thus split, reducing disassembling and assembling time to a minimum. See Fig. 5, right.

In the larger frames another important change has been made. The high-pressure wheel of the Terry type which was incorporated in the first machine has been superseded by a two-row multi-velocity type of wheel running at a high peripheral speed. Extensive experiments have shown that, up to certain peripheral speeds and certain powers with a given thermal drop, the Terry type of bucket is well adapted, but beyond this range the two-row bladed wheel has the advantage. See Fig. 5.

There are several factors entering into the correct design of a wheel of this type other than the actual or theoretical blade efficiency, which make this problem interesting and more complex than one would suppose from a superficial study of the subject on a purely blade-efficiency basis. Disk friction, the power transmitted or rated power of the machine, commercial considerations regarding first cost (which controls the selling price), are all big factors independent of any blade-efficiency theory.

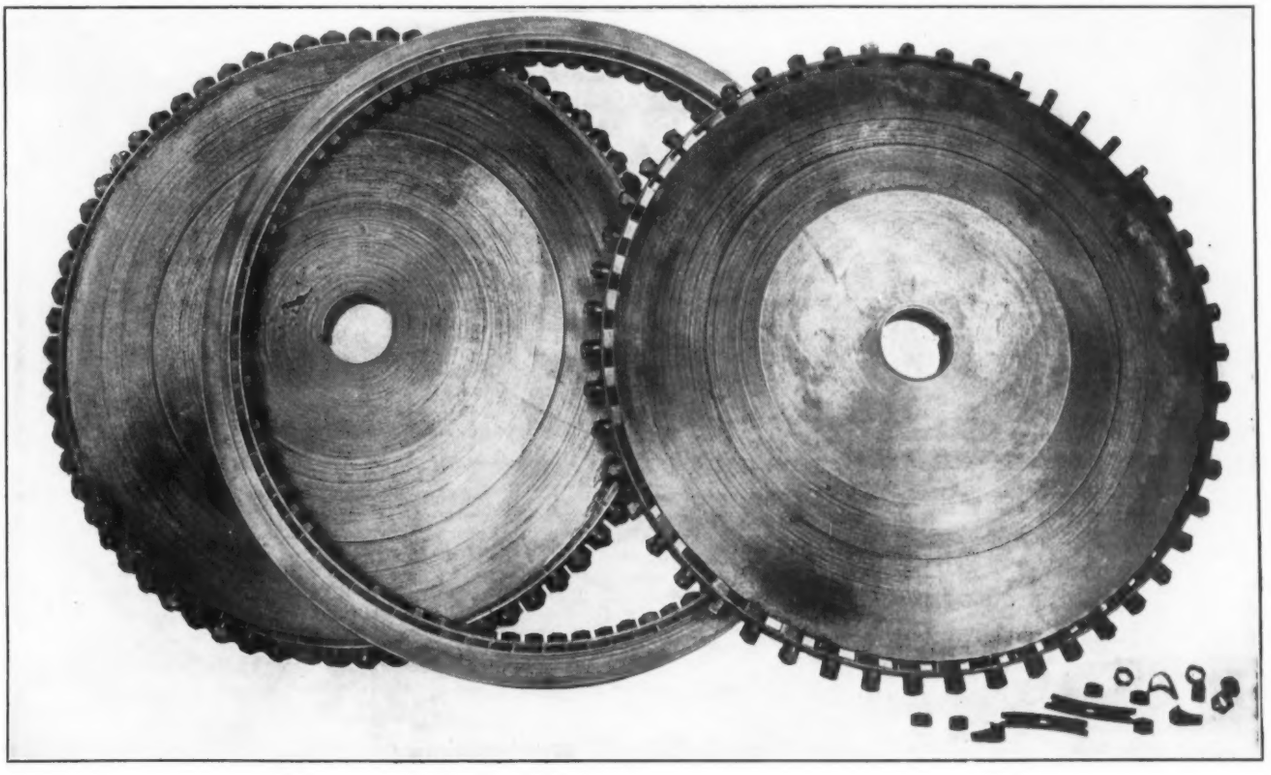


FIG. 3. WHEELS AND GUIDE VANES OF EARLY TERRY TURBINE

The return-flow turbine is designed so that the pressure in the first stage will be about 2 to 5 lb. above the atmosphere. With ordinary steam pressures of, say 150 lb., and allowing two impulses, the peripheral velocity of the buckets must be about 636 ft. per sec. for best efficiency. At 3600 r.p.m. this calls for a pitch diameter of 40½ in. For three reversals the diameter of wheel would be in the neighborhood of 24 to 26 in.

Fig. 1 shows the relative efficiencies of three types of wheels and the important relation that skin friction and windage bear to the overall efficiency. A "two-velocity stage" wheel is more efficient from a blade-efficiency stand-

point than a "three- or four-velocity stage" (see Fig. 2), yet the friction created by the increased diameter is far more undesirable than one would at first imagine, and the advantage gained by augmented blade efficiency is more than offset by the added losses in other directions.

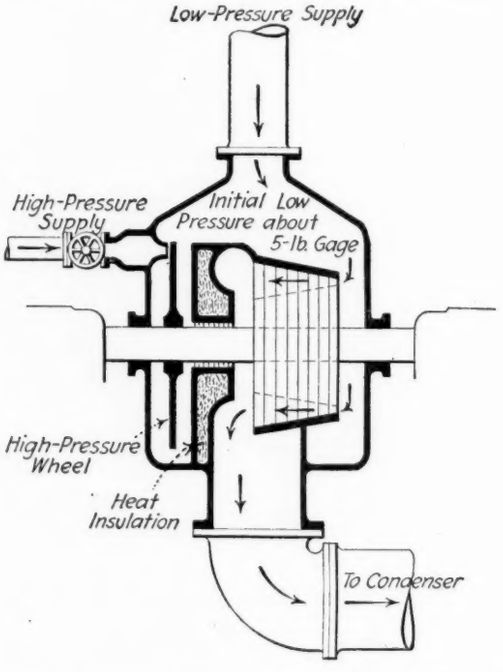



FIG. 4. DIAGRAMMATIC SECTION OF RETURN-FLOW TURBINE

The various formulas of Stodola, Lewicki, Odell and others, for skin friction of disks, show clearly how big a factor this can be, and while these formulas are somewhat vague and indefinite regarding certain conditions that have to be taken into account, they all agree that this friction loss varies as the second to the 2.5 power of the diameter of the wheel and nearly as the cube of the peripheral velocity. Again, these formulas do not take into account the windage of the blading, which is obviously greater in a bladed wheel than on a bucket wheel.

In reverting to the bladed type of wheel in the high-pressure end, instead of the semicircular-type bucket, it is interesting to note in passing that the first turbine experimented with by E. C. Terry in 1893 was on this principle. Fig. 3 shows the wheels and guide blades of this early machine, the patent number of which is 508,-190.

The Terry type of machine is, primarily and essentially, a noncondensing turbine. Its simplicity and consequent unlikelihood of derangement make it an ideal machine for the duties that it is called upon to perform. Within certain limits of speed, vacuum, etc., the two- or three-stage Terry combination makes an equally good condensing machine, having a thermal efficiency as high as that of the single-stage noncondensing design; but when confronted with the necessity for high speeds, high vacuum and larger powers, the wheel becomes impracticable at the low-pressure end owing to its inability to handle a large volume of steam to the best advantage. In the later machines, therefore, the low-pressure wheel has been replaced by a series of single velocity-stage impulse wheels. That practically all authorities agree that this type of wheel for low-pressure work forms the ultimate turbine element is evidenced by the fact that it is being adopted by practically all turbine builders of both large and small machines, with the one exception of the builders of the reaction, or Parsons, type; and that this type of machine is not adaptable to small powers is evidenced by the fact that the builders themselves resort to the impulse principle in their smaller designs.

Again, the "composite design" of velocity staging in the high-pressure end and pressure staging in the low-pressure must be the last word in turbine development if latest designs of practically all the turbine builders both here and in Europe are any criterion. [See the article on page 436 of this issue.—EDITOR.]

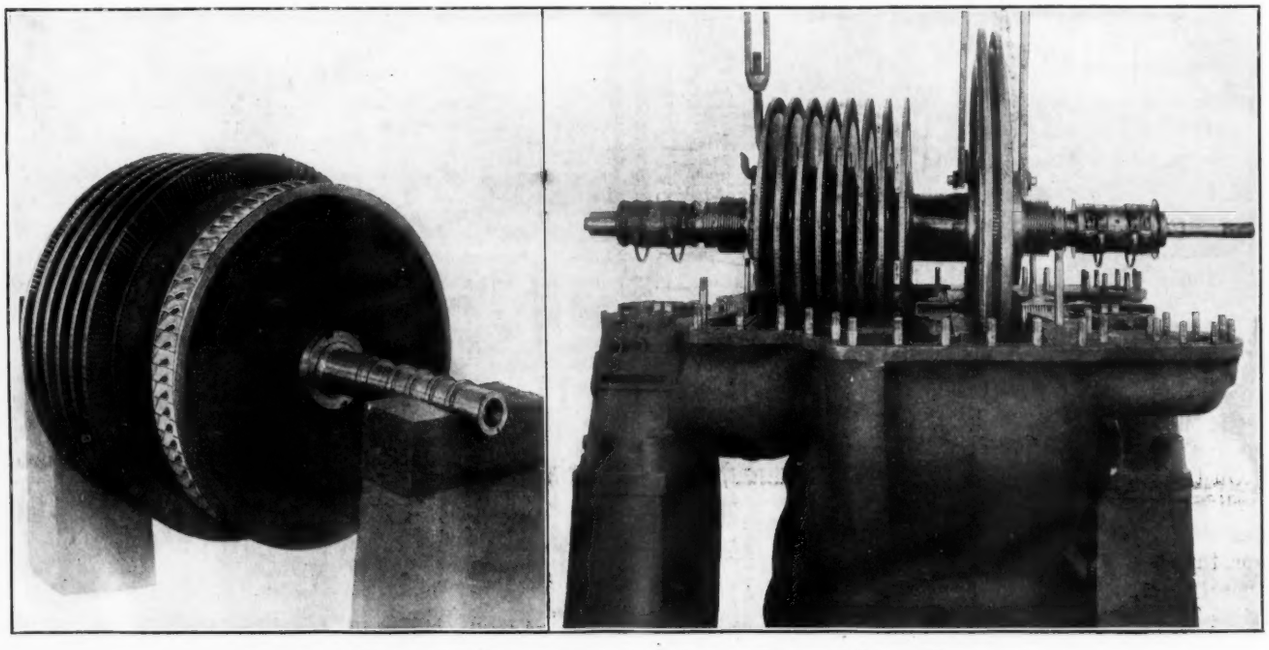


FIG. 5. ROTORS OF OLD AND NEW DESIGNS OF RETURN-FLOW TURBINES

Old design of rotor having Terry high-pressure bucket wheel

Rotor of new machine having two rows of impulse blading on the high-pressure wheel

The obvious advantage of high vacuum in a turbine, particularly in a low-pressure turbine, with the difficulty of designing, building or keeping glands vacuum tight without the necessity of a water seal with its attendant

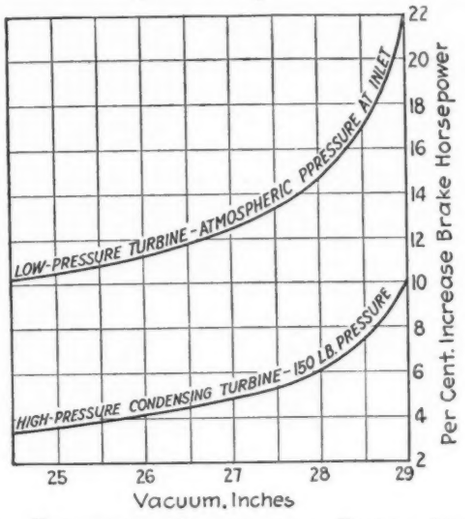



FIG. 6. PERCENTAGE INCREASE IN POWER AVAILABLE (THEORETICAL) PER INCH VACUUM WITH CONDENSING TURBINES

piping and subsequent sediment troubles, led to the departure from the orthodox straight-flow principle to the return-flow design for the elimination of this long-standing bugbear in turbine work. This question of gland leakage often results in trouble between the turbine and the condenser makers when trying to meet guarantees, while the customer looks on and sees the machine run at a lower vacuum than called for and pays the coal bill anyway.

It is often advocated that with a steam seal on the glands and a little steam blowing outward into the engine room there cannot be any air leaking into the turbine. This contention is wrong, as has been demonstrated many times. It often happens that there is a counter current going on, air traveling along one part of the gland and steam passing out of the glands in the opposite direction, this condition being the hardest possible phenomenon to detect. Fig. 4 shows diagrammatically the construction of the return-flow turbine with the low-pressure end in a complete envelope of steam above atmospheric pressure, eliminating the possibility of air leaking into the casing. The rotor and the lower half of the casing of the return-flow machine are shown in Fig. 5.

One other important feature in connection with the arrangement of glands on the return-flow turbine is that no supplementary steam supply is necessary for sealing them when under full load, and even at light loads any steam that finds its way through must pass through the low-pressure end of the turbine, thereby doing work, whereas with the ordinary type of steam-sealed glands all the steam which does manage to escape goes directly to the condenser without doing any further work. That this auxiliary steam supply can amount to quite a factor is evidenced by various tests that have been made. Of course, when a machine is new the glands are tight, so that the leakage during this period is imperceptible, but after setting the machine for commercial operation or if it has been in operation for some time, it is hard to know without repeated tests what this steam leakage amounts to. In big machines this is never, however, a serious amount, but in small ones it can be quite a big percentage of the total steam used.

A careful analytical study of the performance of labyrinth glands was made and published by H. M. Martin, and the formula derived from his experiments is given in his book on steam turbines, as follows:

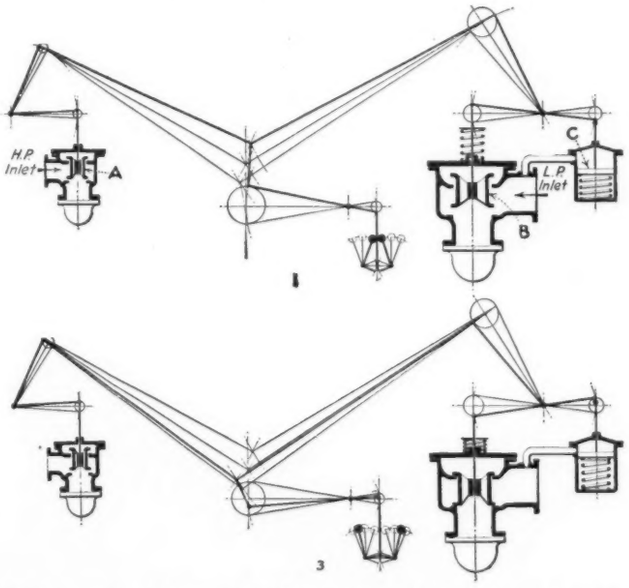



FIG. 7. DIAGRAMS OF LINK MOTION TERRY MIXED-PRESSURE GOVERNOR CONTROL (RATEAU SYSTEM)

1. POSITION when STARTING turbine; high- and low-pressure valves open. Piston C forced down by ample supply of low-pressure steam.
2. SPEED REGULATION: High-pressure valve closed. Turbine running on ample supply of low-pressure steam.
3. PRESSURE REGULATION: Low-pressure supply stopped; piston lifted by spring, closing low-pressure valve and opening high-pressure valve. Governor is always free to close both valves if load is suddenly taken off the turbine.

$$W = 68\,A\sqrt{\frac{P_1\left(1-\frac{l}{x^2}\right)}{V_1\,(N+\log_e x)}}$$

where

$W$ = Weight discharged in pounds per second;
$A$ = Area in square feet available for flow;
$P_1$ = Initial absolute pressure in pounds per square inch;
$V_1$ = Initial specific volume of the steam;
$N$ = Number of points at which the steam is wire drawn;
$x = \frac{P_1}{P_2}$, where $P_2$ denotes the absolute pressure on final discharge from the last ring of the packing.

This formula checks up fairly closely with actual tests made by the writer on a 2000-kw. machine having a mean labyrinth diameter of 8 in. and 12 elements or restrictions.

From the formula it will be seen that the amount of steam passed by a labyrinth gland is directly proportional to the diameter of the glands. This diameter of

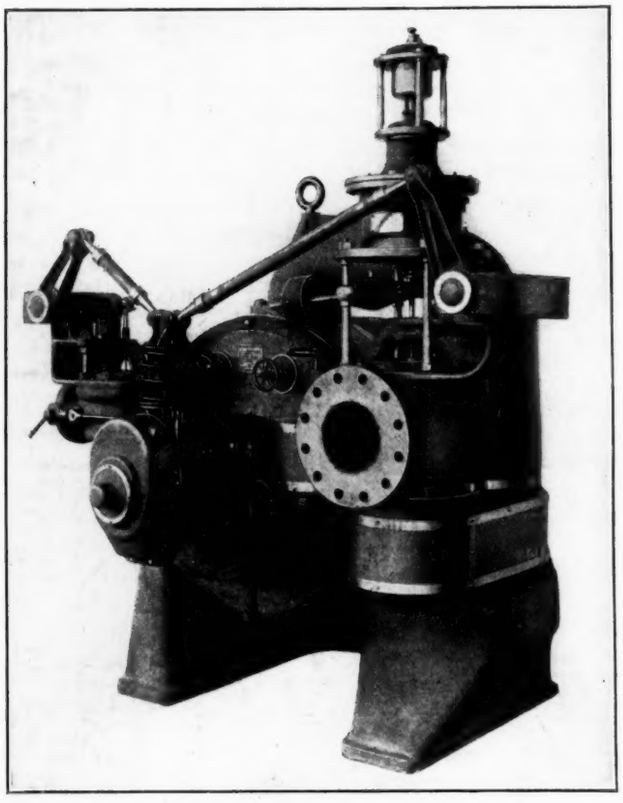


FIG. 8. TERRY RETURN-FLOW TURBINE WITH RATEAU MIXED-PRESSURE CONTROL MECHANISM

gland does not follow any relation to the output of the turbine, and it will be seen that the smaller the machine the larger the percentage of steam that will be passed by the gland; so, as mentioned above, while the amount of steam passed by a labyrinth gland in large machines can be an insignificant factor, it is obvious that in the small machines it can be a serious item. For instance, the figures mentioned above in connection with the 2000-kw. machine show the total steam passed as 177 lb. per hr. On the basis of 15 lb. per kw. this would give a percentage loss due to the glands of 0.6 per cent., whereas reducing this quantity in the ratio of the diameter of the glands, namely, 8 in. to, say 5 in. on a 200-kw. machine, the gland leakage would be reduced to 111 lb., but the percentage of the total steam consumption would be increased to 2.5 per cent., the latter based on a water rate for the smaller machines of 22 lb. per kw.-hr.

Fig. 6 shows the theoretical saving per inch of vacuum in a straight high-pressure condensing and a low-pressure turbine. In the low-pressure machine the vacuum is, therefore, of much importance. We must not look upon this as a question of efficiency so much as a question of how much horsepower one can obtain from a given amount of exhaust steam. Then it means that increasing the vacuum from 27 in. to 28 in. the amount of power available from a fixed quantity of exhaust steam is increased 15 per cent. No precaution is too great for the purchaser to take to insure himself against losses at this point irrespective of any guarantees that may be given either from the turbine or the condenser builder.

The success of the low-pressure turbine intelligently installed is undisputed, but the bulk of the research and development work has been along the lines of the larger machines. The marked saving in these machines has naturally led to the introduction of the low-pressure turbine in small plants such as breweries, ice plants, etc., with just as successful results as with the larger units. Low-pressure turbines of 50 hp. and more have recently been installed, and many more installations are in course of construction. With the exception of a few isolated cases where a fixed supply of exhaust steam can be depended upon indefinitely, the low-pressure turbine has given place to the mixed-pressure machine, the latter having the advantage that should anything happen to the engine or other source of low-pressure steam supply, the full power of the turbine can be obtained when operated with high- or mixed-pressure steam. The return-flow turbine is particularly applicable to low- and mixed-pressure work, as the effect of vacuum in a machine of this kind is of much more importance than in a high-pressure condensing machine.

For satisfactory operation under mixed-pressure conditions and where the low-pressure supply is liable to decrease or fail, a special arrangement of governor mechanism is designed, so that the high-pressure steam is automatically admitted to make up for any deficiency in the low-pressure supply. The system employed by the Terry Steam Turbine Co. on its mixed-pressure turbines is what is known as the mixed-pressure Rateau control, manufactured under license from the Rateau Steam Regenerator Co., the design being modified to eliminate the complicated oil-relay mechanism necessary on larger machines.

The direct-connected governor of the Terry type is used directly connected to the governor valves. The

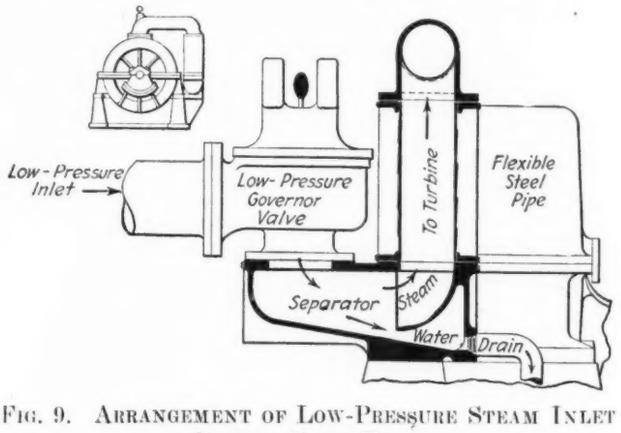



FIG. 9. ARRANGEMENT OF LOW-PRESSURE STEAM INLET ON RETURN-FLOW TURBINE

principle of this mechanism is shown in Fig. 7, and a photograph of the actual machine fitted with this control is shown in Fig. 8. All the levers are mounted on ball bearings to eliminate friction as far as possible. The governor running at high speed has more power than the usual low-speed geared governor and the mechanism itself being balanced by counterweights as

shown in Fig. 8, the governor is relieved of all external loads other than to operate the balanced valves.

In all turbine practice, both in high- and low-pressure machines, the question of pipe connections and the elimination of stresses on the turbine is well known to be a serious problem. This is particularly so with the big piping necessary for low-pressure machines, and it is again of further importance in the smaller units, which, on account of their compact size and light weight are susceptible to distortion from outside stresses. The common practice is to bolt the low-pressure valve and piping directly to the turbine casing. This entails considerable risk of pulling the turbine and, consequently, the whole unit out of line, causing vibrations and generally unsatisfactory running. To eliminate this in the return-flow turbine the low-pressure steam supply is not rigidly connected with the turbine proper, but is bolted to an entrainer or separator which in itself is bolted rigidly to the bedplate (see Fig. 9). From this entrainer the steam is led vertically through a flexible steel pipe which leads to the top of the turbine casing. In this way no outside stresses due to the heavy low-pressure steam-supply piping are thrown on the turbine itself. Furthermore, by the introduction of this additional entrainer, drier steam is obtainable in the turbine than would be the case were the inlet piping connected directly to the turbine proper.

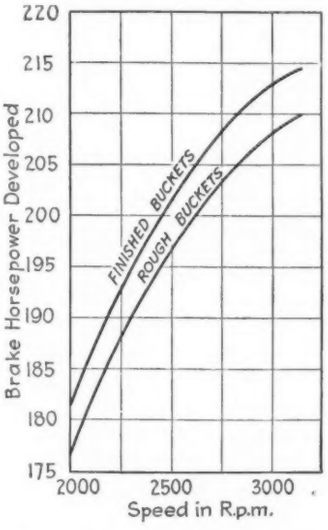



FIG. 10. TESTS OF IMPULSE WHEELS, SHOWING EFFECT OF FINISHING BLADES

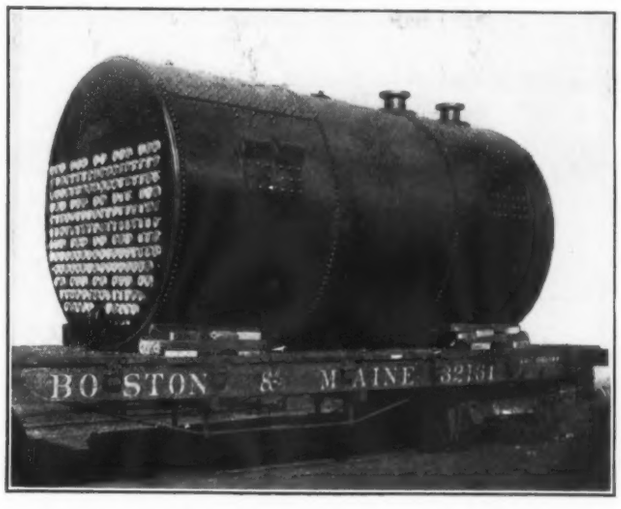
Throughout the design of the machine special attention was paid to obtaining the best possible water rate consistent with a compact and reliable mechanical design, and the figures given in the table of some recent tests show some remarkably high efficiencies for turbines of such small power.

The last degree of efficiency can be obtained only by a careful study of the correct areas through the blade passages and by special attention to the finish of the blading in the wheels to eliminate friction and eddies. By the adoption of drawn material for the blading, true areas can be obtained. This machine, as built, conforms closely to calculations, as has been evidenced repeatedly by the careful observation of pressure drops through the various stages, these falling almost exactly in line with the calculated drops. The adoption of polished wheels further enhances the efficiency.

Some interesting experiments were carried out to determine the effect of rough and finished blades and wheels. Fig. 10 shows two curves, one with rough drop-forged blades, the other with blades polished to a true section and with the wheel polished.

## Nine-Foot Return-Tubular Boiler

The first 108-in. diameter return-tubular boiler recorded under the laws of the Commonwealth of Massachusetts was recently built at the shops of the D. M.




NINE-FOOT RETURN-TUBULAR BOILER READY FOR SHIPMENT

PERFORMANCES OF TERRY RETURN-FLOW TURBINES

| Turbine Number | Initial Steam Pres. before Gov. Valve, Lb. | Vac. Exhaust, In., Hg. Referred to 30-In. Barometer | Load in B.hp. | Load in Kw. | Speed in R.p.m. | Initial Steam Condition, Deg. F. Superheat or Quality | Total Steam per Hr. Actual, Lb. | Total Steam per Hr. Corrected to Dry Steam, Lb. | Water Rate per Kw.-hr. Actual, Lb. | Water Rate per B.hp.-hr. Actual, Lb. | Water per B.hp.-hr. Corrected to Dry Steam, Lb. | Thermal Efficiency Ratio | Water Rate on Which Efficiency Ratio Is Based | Test |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1383 | 128.8 | 25.64 | 368.1 | .... | 3640 | 10.6 | 6297.8 | 6376 | .... | 17.10 | 17.29 | 0.535 | Dry steam | A |
| 1383 | 128.8 | 28.00 | 441.3 | .... | 3340 | Dry | .... | 6370 | .... | .... | 14.43 | 0.561 | Dry steam | B |
| 1383 | 128.8 | 24.16 | 254.5 | .... | 3340 | 10.0 | 5016.0 | 5069 | .... | 19.7 | 19.90 | 0.491 | Dry steam | C |
| 1383 | 128.8 | 28.00 | 343.0 | .... | 3640 | Dry | .... | 5069 | .... | .... | 14.77 | 0.548 | Dry steam | D |
| 1383 | 7.03 | 25.60 | 307.0 | .... | 3600 | 85.0 | 8086.0 | .... | .... | 26.35 | .... | 0.606 | Actual | E |
| 1383 | 5.20 | 23.75 | 193.0 | .... | 2000 | 89.0 | 7076.0 | .... | .... | 36.68 | .... | 0.529 | Actual | F |
| 1383 | 2.46 | 21.75 | 205.5 | .... | 3999 | 93.7 | 6063.0 | .... | .... | 29.48 | .... | 0.619 | Actual | G |
| 1383 | 8.25 | 24.35 | 307.5 | .... | 3300 | 76.5 | 8340.0 | .... | .... | 27.10 | .... | 0.639 | Actual | H |
| 1874 | 1.75 | 25.30 | .... | 106.4 | 2500 | 12.0 | 6220.0 | .... | 58.5 | 38.4 | .... | 0.521 | Actual | I |
| 1874 | 100.4 | 25.8 | .... | 98.8 | 2462 | 0.981 | 3440.0 | 3310 | 34.85 | 22.85 | 22.00 | 0.443 | Dry steam | J |
| 1874 | 100.0 | 0.0 | .... | 98.2 | 2525 | 12.5 | 6065.0 | 6142 | 61.8 | 40.58 | 41.10 | 0.414 | Dry steam | K |
| 1750 | 125.5 | 25.25 | 416.9 | .... | 3600 | 26.9 | 6400.0 | 6575 | .... | 15.35 | 15.78 | 0.597 | Dry steam | L |
| 1750 | 125.5 | 28.00 | 490.9 | .... | 3600 | Dry | .... | 6575 | .... | .... | 13.39 | 0.609 | Dry steam | M |
| 1750 | 125.0 | 26.57 | 321.7 | .... | 3600 | 21.0 | 5133.0 | 5215 | .... | 16.30 | 15.96 | 0.558 | Dry steam | N |
| 1750 | 125.0 | 27.36 | 233.4 | .... | 3600 | 25.0 | 3774.6 | 3870 | .... | 16.17 | 16.59 | 0.512 | Dry steam | O |
| 1750 | 125.5 | 25.25 | 419.6 | .... | 3200 | 26.9 | 6400.0 | 6575 | .... | 15.26 | 15.67 | 0.604 | Dry steam | P |
| 1750 | 125.5 | 25.25 | 407.6 | .... | 2800 | 26.9 | 6400.0 | 6575 | .... | 15.71 | 16.13 | 0.587 | Dry steam | Q |
| 1750 | 125.5 | 25.25 | 384.9 | .... | 2400 | 26.9 | 6400.0 | 6575 | .... | 16.63 | 17.08 | 0.554 | Dry steam | R |
| 1750 | 0.25 | 26.01 | 67.0 | .... | 500 | 110.0 | 6874.0 | .... | .... | 102.6 | .... | 0.175 | Actual | S |
| 1750 | 0.25 | 26.01 | 167.5 | .... | 1500 | 110.0 | 6874.0 | .... | .... | 41.04 | .... | 0.438 | Actual | T |
| 1750 | 0.25 | 26.01 | 237.6 | .... | 3500 | 110.0 | 6874.0 | .... | .... | 28.94 | .... | 0.620 | Actual | U |
| 1750 | 0.25 | 26.01 | 227.8 | .... | 4000 | 110.0 | 6874.0 | .... | .... | 30.19 | .... | 0.595 | Actual | V |
| 2044 | 124.7 | 26.49 | 343.0 | .... | 3600 | 27.4 | 5165.0 | 5305 | .... | 15.06 | 15.47 | 0.578 | Dry steam | W |
| 2044 | 125.0 | 26.49 | 295.0 | .... | 3600 | Dry | .... | 4665 | .... | .... | 15.82 | 0.566 | Dry steam | X |
| 2044 | 125.0 | 28.00 | 325.0 | .... | 3600 | Dry | .... | 4665 | .... | .... | 14.36 | 0.568 | Dry steam | Y |
| 2044 | 125.0 | 26.49 | 240.0 | .... | 3600 | Dry | .... | 3990 | .... | .... | 16.62 | 0.539 | Dry steam | Z |
| 2044 | 125.0 | 26.49 | 180.0 | .... | 3600 | Dry | .... | 3260 | .... | .... | 18.10 | 0.495 | Dry steam | AA |

Dillon Steam Boiler Works, Fitchburg, in accordance with the Massachusetts state laws, for a safe working pressure of 125 lb. The recorded number given to it at the State House in Boston is 971.

The boiler is 9 ft. in diameter and 18 ft. long. The firebox-steel shell plates, and the flange-steel heads are ⅝ in. thick. The boiler has 200 4-in. by 18-ft. Parkesburg charcoal-iron tubes, and has a butt-strap, double-riveted steam drum 30 in. diameter by 6 ft. long. The bare boiler without tubes weighs 27,000 lb. (13½ tons) and complete with all castings and fittings, 84,440 lb. (42.22 tons).

In order to make the boiler lighter for transportation but sixteen tubes were put in at the time of shipment. These were to brace the tube sheet, the rest of the holes being plugged so that the boiler would not sink in case it fell overboard while unloading it from the vessel to the lighter, or in case it was necessary to float it from the boat to the shore. It was shipped to Porto Rico.

The boiler is of interest on account of its large diameter, and because it is the first of its size recorded under the laws of the Commonwealth of Massachusetts.

# Metallurgical and Special Furnaces*

By Osborn Monnett†

*SYNOPSIS—Industrial furnaces, hand-fired or stoker equipped, specially designed to eliminate smoke.*

In studying the smoke problem it will be noticed that there are a great many furnaces burning coal and making smoke other than those installed under a boiler. In the various industries common to a large city enormous quantities of coal are consumed in metallurgical and special furnaces which have no application whatever to the generation of steam. The nature of the work demanded of these furnaces is such as to make them peculiarly susceptible to the making of smoke; in fact, some of the processes require such a low rate of combustion that it is difficult to get the proper temperature for complete combustion. In this class one of the worst smokers has been the ordinary hand-fired annealing oven.

On a large scale the use of powdered coal has successfully cleaned up this class of plant and has generally resulted in a substantial saving of fuel over the hand firing, although the investment required is considerable. On a smaller scale, or where the product is of small size, such as automobile parts, producer gas has been successfully used. Producer gas is also well adapted for enameling ovens, and china, pottery and terra cotta kilns.

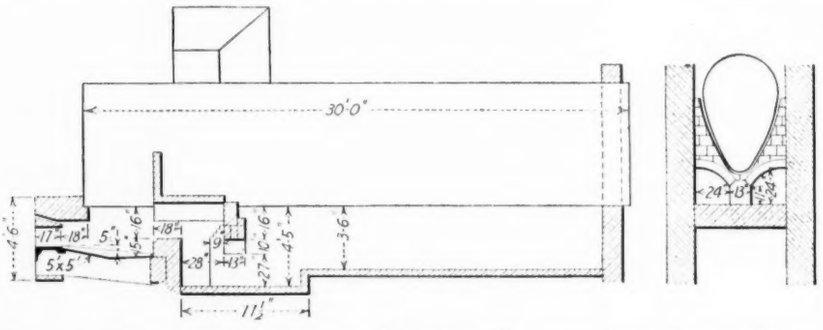



Fig. 1. Fertilizer Tank with Modified Double-Arch Bridge-Wall Setting

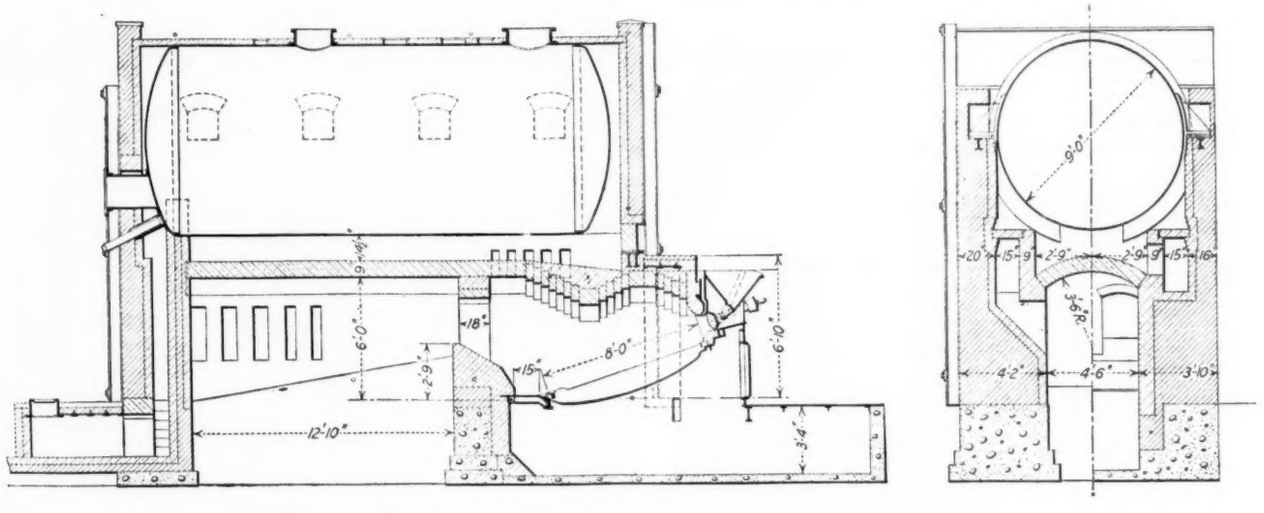



Fig. 2. Crude-Oil Still with Hand-Fired Coking Furnace

Fig. 1 shows the layout of a hand-fired low-temperature furnace in connection with a fertilizer tank. The setting, which is of the double-arch bridge-wall design, is low, and excavation has been made under the furnace proper. Using the coking method of firing and ample air admission

*Copyright, 1915, by Osborn Monnett.
†Smoke inspector, City of Chicago.

for the semibituminous coal, this furnace can be operated without dense smoke, but it requires careful attention and this is difficult to get in this class of plant.

A crude-oil still, with a special hand-fired furnace arranged for low rates of combustion, is shown in Fig. 2. This furnace is designed for the coking method of firing with semibituminous coal. There is considerable brickwork in the furnace, designed not only to give good mixture, but also to isolate the shell from the heat to prevent burning of the still. Too high a temperature at any one point would be disastrous, so the furnace is provided with a spring arch the entire length of the tank. Coal is charged on the front of the grate and pushed down when fully coked.

Natural gas and fuel oil are also used for this and similar work where the price is low enough so that these fuels may compete favorably with coal. When burning the latter any of the various gravity-feed furnaces can be used. Fig. 3, showing a small Burke furnace attached to a pot annealing oven, is typical of this class of service. Powdered coal and producer gas are also being used in this work. There is probably no class of metallurgical work in which one or the other of the above fuels will not give satisfaction.

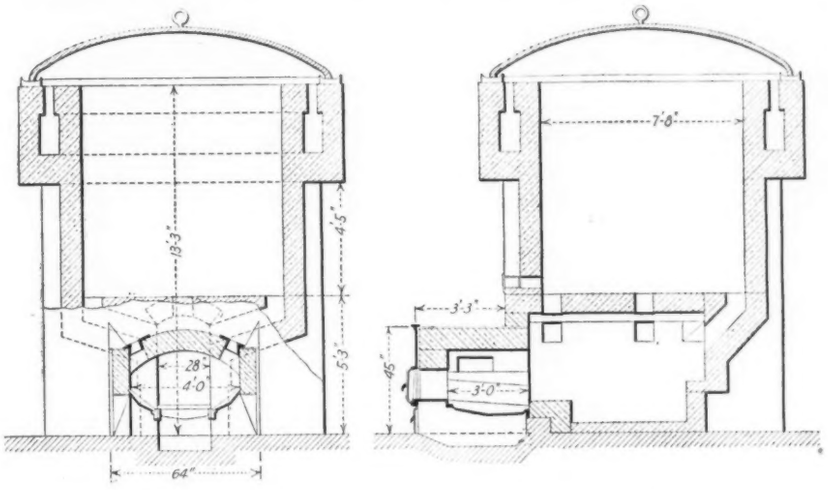



FIG. 3. POT ANNEALING OVEN AND BURKE GRAVITY-FEED FURNACE

As mentioned before, the underfeed stoker is peculiarly adapted to special furnace work, owing to the fact that the necessary air for combustion is supplied by mechanical means. Figs. 4 and 5 show typical reheating furnaces equipped with this type of stoker. Rotary drying is another service in which the underfeed stoker works out well. Natural draft on this work is an uncertain quantity, placing at a disadvantage stokers depending on a stack for their air supply. A furnace arranged for connection to a rotary dryer is shown in Fig. 6. This outfit is adapted to the drying of fertilizer, garbage, blood, sand, sugar-beet pulp, or any similar substance from which it is desired to drive off the moisture.

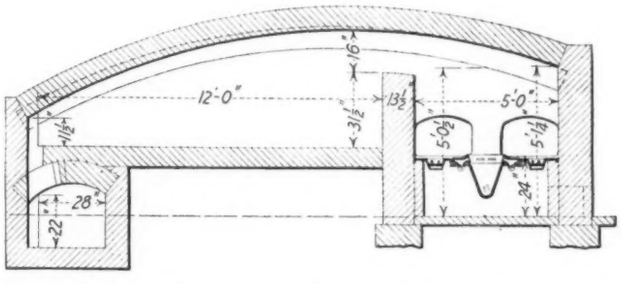



FIG. 4. JONES UNDERFEED STOKER SERVING REHEATING FURNACE WITH UNDERGROUND BREECHING

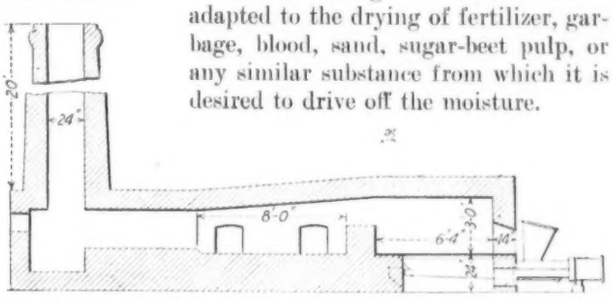



FIG. 6. ROTARY DRYING FURNACE FITTED WITH UNDERFEED STOKER

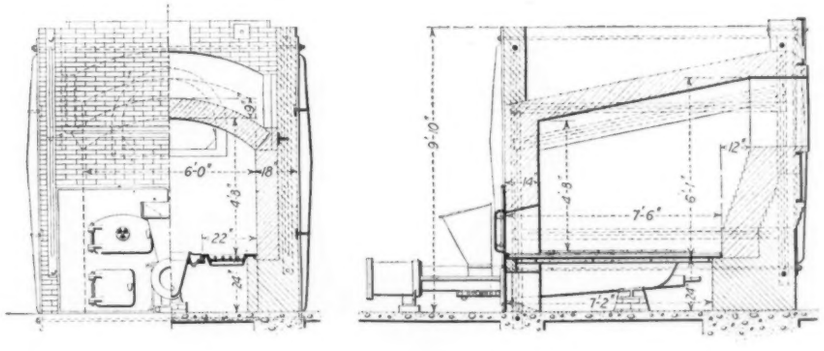



FIG. 5. TYPICAL REHEATING FURNACE WITH UNDERFEED STOKER AND INDEPENDENT STACK

**Ocean Volume to Land Area**—One per cent. of the contents of the oceans would cover all the land areas of the globe to a depth of 290 ft.—U. S. Geological Survey.

**Horsepower and Torque Defined**—It is important to understand clearly the difference between horsepower and torque. The former is the rate of doing work, while the latter is only one of the quantities making up horsepower. The torque of a motor is sometimes defined as the pull or force exerted at the surface of the armature, multiplied by the radius of the armature. For commercial purposes, however, is is defined as the pull exerted at a certain radius from the shaft center. For convenience this pull is usually expressed in pounds and the radius in feet, which, multiplied by the peripheral speed, gives an expression in foot-pounds which is readily reducible to horsepower. Assuming that the force is applied at a distance of one foot from the center of the shaft so that r (radius) = 1, then

$$\text{torque (T)} = \frac{\text{hp.} \times 33{,}000}{2\pi \times \text{r.p.m.}} = \frac{\text{hp.} \times 5252}{\text{r.p.m.}}$$

From this it is evident that for a given motor and a given horsepower, the torque varies inversely as the speed. If the first definition of torque is assumed, that is, the force acting at the surface of the armature—it is apparent that the torque would be dependent on the diameter of the armature as well as the speed; whereas, by the second definition, it is independent of the armature diameter. In motor applications one is concerned with the torque exerted on the motor shaft and not at the surface of the armature.

# High-Tension Switching Systems

By John A. Randolph

*SYNOPSIS—Factors determining the extent and arrangement of high-tension switching equipment to be employed, and descriptions of several of the more common systems in use.*

In determining the arrangement of a high-tension switching system several fundamental factors must be considered. One is the nature of the service, inasmuch as the scheme of connections for a lighting and industrial service is generally different from that of a system feeding railways. Another is the type of station (central station or substation), which will largely determine certain features of the design. The magnitude and extent of service will also have an important bearing, as will the number of high-tension feeders and the voltage to be carried. The distance of the station from its center of distribution is a vital factor, as is also the space available for installation purposes. Safety to life and property should not be overlooked, although this is perhaps of more importance in the design of the compartments and supports than in the determination of the diagrammatic layout. Furthermore, the money available for construction purposes is a governing factor, and precautions must be taken to insure continuity of service at all times.

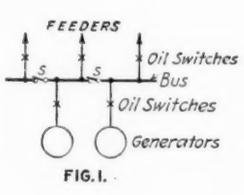


FIG. 1.

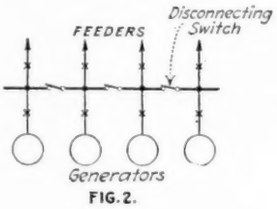


FIG. 2.

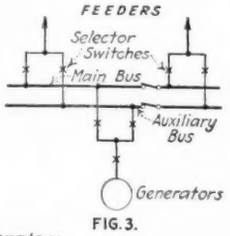


FIG. 3.

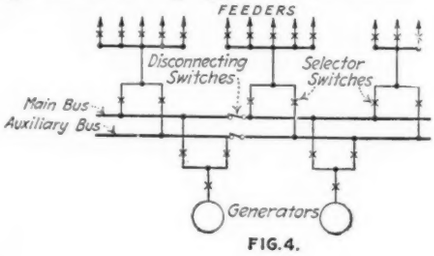


FIG. 4.

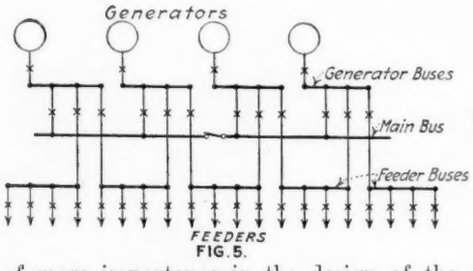


FIG. 5.

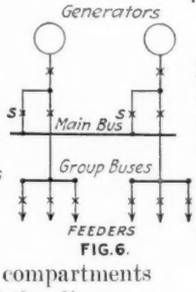


FIG. 6.

In regard to the nature of the general service, it can be said of the railway system that, owing to the large amount of exposed conductors, such as third rails, trolley wires, underground contact rails and ground returns, the liability to short-circuits and consequent interruptions in operation is somewhat greater than on lighting systems, especially such of the latter as are installed in underground conduits. Therefore, greater precautions against shutdowns are advisable for the railway station.

The number of high-tension feeders and the voltage carried will depend largely upon the extent of the service and the distance of the source of supply from the distribution center. If the generating station, for instance, is run by hydro-electric power and is located in the mountains a long distance from the point at which the energy is used, it will be necessary to make the number of high-tension feeders as low as possible in order to save transmission-line costs. To do this the line voltage must be raised in order to transmit the given power over the limited cross-section of conductor. This will permit a comparatively small number of high-tension switches being used, but they will be larger and more cumbersome than those for lower voltages. Wide clearances between conductors and between live parts and ground must be maintained in extra high-voltage work; therefore, more space is required for a given amount of apparatus and conductors than in the case of lower-voltage installations. If this space is limited it may be necessary to limit the output of the station. Here also, the factor of safety to life and property is of more importance than in other cases.

The money available for construction purposes is a determining factor in that it may be necessary to sacrifice many advantageous features to save expense. This is often hazardous, but exigencies may demand it. To conform to this restriction it may be necessary to use but one bus, with a correspondingly smaller number of switches, or to otherwise arrange the switching apparatus in the simplest possible manner, which is by no means the safest. However, in connection with the first cost, the future continuity of service should be borne in mind. Lack of patronage, due to unreliable service, may cause the company to lose in a comparatively short time an amount greater than the saving effected by limiting the flexibility of the switching system.

## Single-Bus System

The simplest method generally employed for the arrangement of switches and buses on a high-tension system is shown in Fig. 1 (the "bus" in this case signifying one set of busbars). One bus is used to which all the generators and feeders are directly connected, there being only one oil switch to each generator and to each feeder circuit. Disconnecting switches *S* are generally installed for isolating sections on which serious defects exist or on which it is desired to make repairs. This arrangement is often used in railway substations. To add to the flexibility of such a plan, the machines and feeders are sometimes connected to the busbars at the same points, disconnecting switches being placed between these points, as in Fig. 2. This provides better facilities for the isolation of feeders and machines with their corresponding bus sections.

The single-bus system has the advantage of minimum space and comparatively low initial cost, but it has the disadvantage of not making possible the ready transfer of machines or feeders from the sections to which they are normally connected, to other sections. On a single-

bus system such as shown in Figs. 1 and 2, the machines and feeders always receive or deliver their energy at a fixed point. Moreover, if a sectionalizing switch is opened, the sections on either side of the gap are entirely separated and work independently.

### Two-Bus System

In the larger modern central stations, it is customary to install two buses—the main bus and the auxiliary bus. There are several methods by which the feeders and generators may be connected to the buses. In one of these, shown in Fig. 3, two selector switches with each feeder and alternator permit a connection to either of the two buses. Sectionalizing switches are also located at intervals in each bus. With this arrangement, if it becomes necessary to isolate a section of either bus the feeders and machines previously connected to that section can be easily transferred to the other bus without interruption in the service. Moreover, if the demand on one bus becomes so heavy that it is in danger of becoming overloaded, part of the load can be transferred to the other bus; or, where two buses are used, it is possible to so divide the load on the station that accidents occurring on one part of the system will not affect the system as a whole.

### Group System

An arrangement commonly employed in central stations and on railway systems using a number of substations for distribution is shown in Fig. 4. In this scheme a main and an auxiliary bus are used, but the feeders, instead of having independent connections to the two operating buses, are connected to group buses which, in turn, are connected by selector switches to the main and auxiliary buses. An advantage of this arrangement is that fewer selector switches are necessary; also, in cases of emergency, a number of feeders can be transferred simultaneously from one operating bus to the other, thus saving time and simplifying the work of the operator.

Another diagram making use of a group bus for feeders is shown in Fig. 5. In addition to the main bus and the feeder group buses, there is a series of generator buses. Four circuits are connected to each generator bus section—one comprising the feed from the generator or other source of power, another a connection to the main bus, and the other two serving to connect the generator bus to two adjacent feeder group buses. The function of the main bus is to tie all the individual buses together and to maintain the power supply on any generator bus after its particular machine has been shut down. Furthermore, it serves to equalize the load on all the generators. The outgoing feeders are connected to the group buses. Inasmuch as every group bus is connected to two adjacent generator buses, it is possible to isolate a generator or group bus without serious inconvenience to the system as a whole. Moreover, all or a part of the main bus can be disconnected from the rest of the system without interrupting the service. Another advantage is that one or two groups of feeders can be fed from either one or two generators independently of the rest of the system. This affords especial convenience in testing.

An arrangement somewhat similar to that of Fig. 5, but incapable of as many combinations, is shown in Fig. 6. A main operating bus is used whereby all the generators and all the feeder groups, or both, may be tied together through the main bus switches *S*. The generators may be disconnected at will as long as the main bus is kept alive and the switches *S* are closed. Moreover, the individual feeder groups may be isolated from the main bus as long as the respective generators for those groups are kept running. A disadvantage of this system is that an isolated group, if kept alive, is absolutely dependent upon one particular generator. Furthermore, if the main feed line of the group is out of commission the group itself is also put out of service. The same result will ensue if the generator bus is out of service. In the more flexible arrangement of Fig. 5 these disadvantages do not exist, inasmuch as every group has a connection to two separate generator buses.

### Multiple-Voltage System

In the systems described thus far the voltage on the buses has been the same as that of the generators. However, where two or more services with different voltages are to be supplied from one central station, it is often

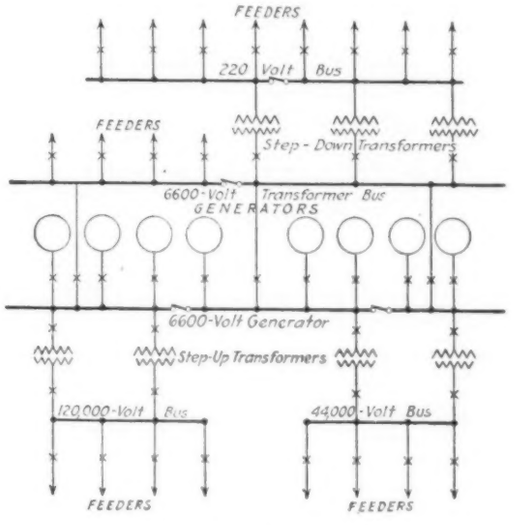



Fig. 7.

advisable to install a separate bus for each service; such an arrangement is illustrated in Fig. 7. The generator voltage of 6600 determines the pressure of the generator bus. However, for the 10,000- and 120,000-volt services, separate buses are installed whose power supply comes from the generator bus, but with the voltage increased by means of step-up transformers. A 220-volt bus for low-tension work is fed through step-down transformers whose primaries are joined to a 6600-volt bus fed directly from the generator bus. For flexibility the various buses are divided into sections, and between these sections are switches which can be quickly opened or closed at the will of the operator.

**The Expansion of Steam Lines** may be found by the formula

$$E = C \times (T - t) \times L$$

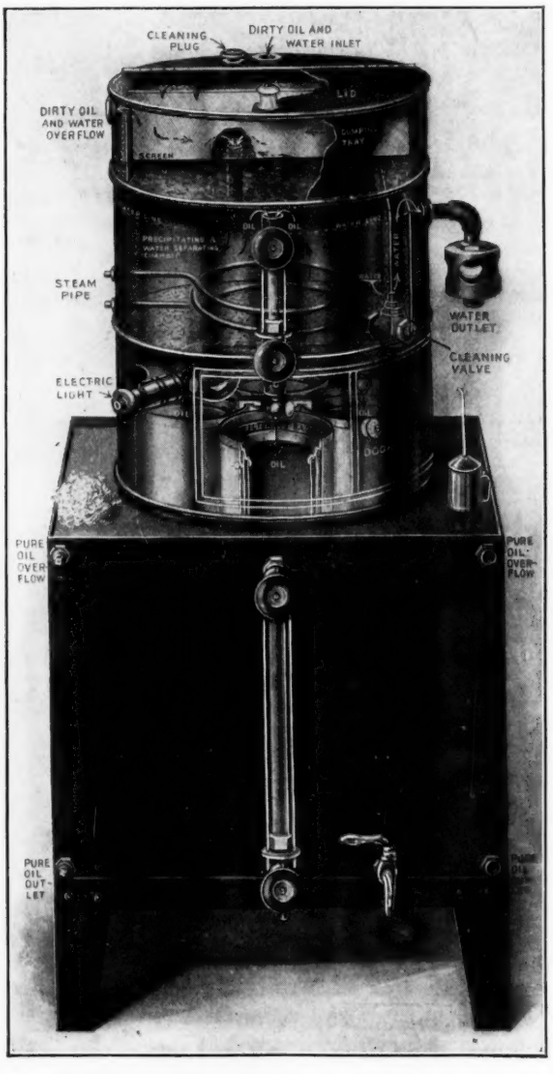
when

E = Expansion in inches;
T — t = Temperature difference;
L = Length of pipe in inches;
C = The coefficient of expansion of the metal of which the pipe is made.

The coefficients of expansion of various pipe materials are: Cast iron, 0.0000065; steel, 0.0000067; wrought iron, 0.0000068, copper, 0.0000095; brass, 0.0000105.

## Nugent Improved Oil Filter

The accompanying illustration shows one of the standard filters now being made by Wm. W. Nugent & Co., of Chicago. This particular design may be circular or square in section. It is made in eight different sizes, the capacities being from 6 to 100 gal. per hr., respectively. Multiplication of the filtering chambers will, of course, give an apparatus of any desired capacity. The cylindrical filtering and water-separating section, shown at the top in the illustration, is independent of the reservoir, and is supplied separately if desired. A tank suitable for oil storage may be available at the plant and if appearances do not count an oil barrel may be used. In the latter case the barrel is turned upright and the filtering chamber set over the open end.




NUGENT IMPROVED OIL FILTER

From the phantom view the path of the oil from inlet to storage may be traced. In the top part are two semicircular sections, one known as the dumping tray and the rear half as the screen chamber. If the filter is connected to an oiling system, the dirty oil and water which may be present enter the dumping tray through the inlet shown. By raising the lid it may also be poured into this compartment. In the vertical wall separating the two sections is a fine-mesh copper screen through which the oil passes into the screen chamber. From here it flows downward through a short pipe into the water-separating and precipitating chamber. It enters under the surface of the water, rises to the top and overflows into a central pipe supplying the filtering bags. The water passes under the inner partition, shown at the right of the chamber, and over the second partition to the outlet.

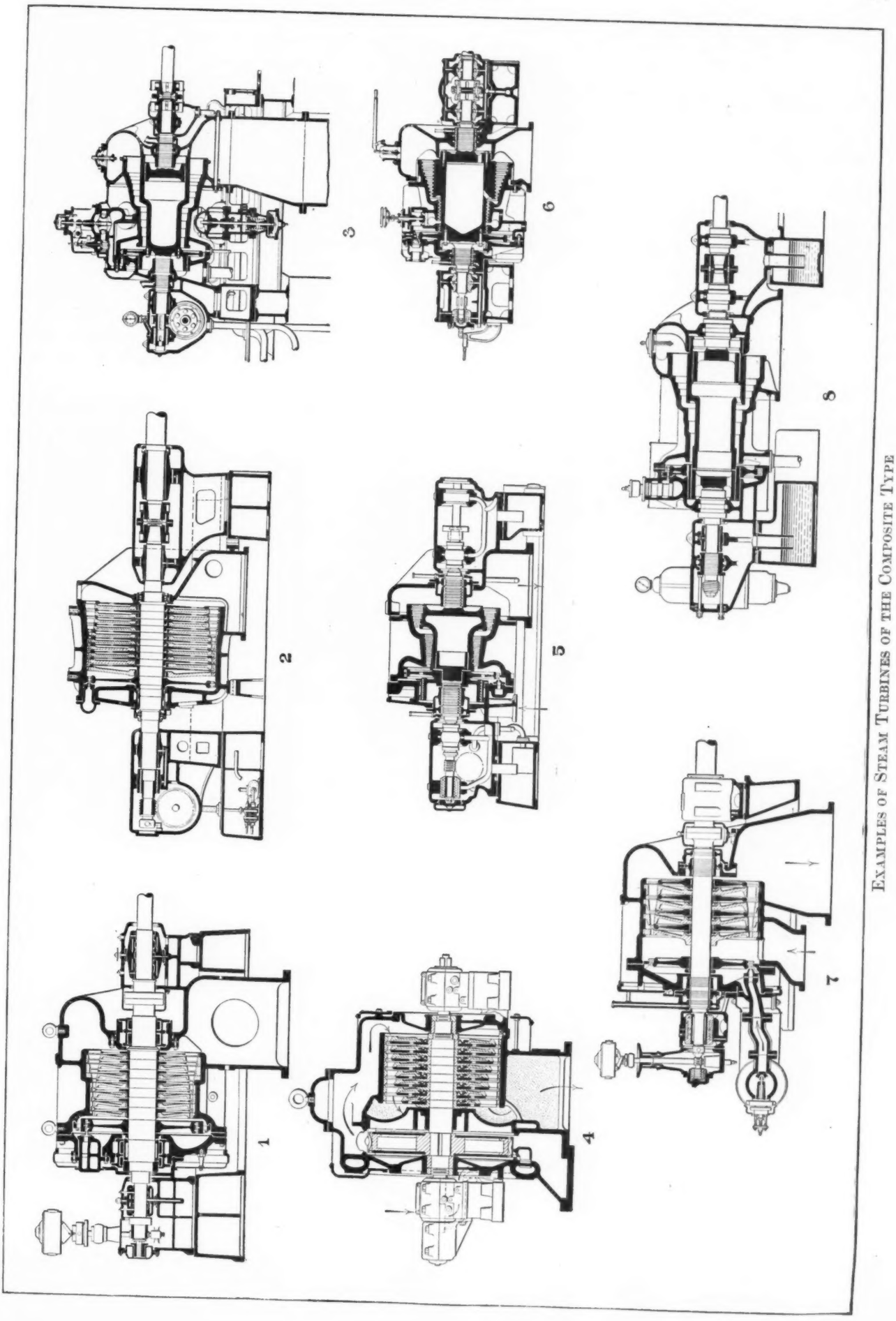
Depending on the size of the filter, one, two, three, four, six or eight sets of filtering bags are provided, three in a set, all on independent rings. The two smaller ones are made of comparatively thin fabric, while the material forming the largest is much heavier. An electric light has been provided so that the filters may be inspected through the sliding door shown in the illustration. The machines having four, six or eight sets of bags are suspended from a central spindle and each may be rotated to a position in front of the door. The cock controlling the drip at this point may be closed and the bags inspected or cleaned. The pipe leading from the screen chamber is of such a size that it cannot supply enough oil to flood the filters. If more should come to the dumping tray than can be cared for, the surplus escapes through the overflow at the top and eventually is returned to the filter. Other features are the steam coil in the water-separating chamber and the facilities for cleaning.

## The Composite-Type Steam Turbine

Five years ago we called attention to the evident tendency to unification in turbine types, and the adoption by several builders of the velocity-stage for the initial expansion, and by the builders of velocity-stage turbines of single-velocity impulse or reaction stages for the lower ranges. We pointed out that by allowing the initial expansion to take place in a single set of nozzles, and using a velocity-stage of two rows of moving and a single row of reversing buckets to absorb the velocity so generated, the steam is reduced in pressure from, say 150, to 30 lb. before it is introduced to the turbine case, decreasing the pressure upon the balancing plates or dummies, diminishing the temperature to which the rotor drum and shell are subjected, and eliminating the long section of short blades, admittedly the least efficient of the reaction turbine, on account of their large windage in the dense medium of the high-pressure steam, and of the large proportion of their clearance to the active surface.

The anticipations expressed in this article have been realized, and we present upon the opposite page a number of the machines of this composite type as now built by prominent companies. The presentation is, however, by no means complete, and should include the General Electric Co. and the Westinghouse Machine Co., if not other American builders. The builders of the turbines, sections of which are shown, are as follows:

1. Bergmann, Berlin.
2. British Westinghouse, Manchester.
3. Melins & Pfenninger, Berlin and Munich.
4. Terry, Hartford, Conn.
5. Gutehoffnungschütte, Oberhausen.
6. Brush, London.
7. Allgemeine Elektricitäts Gesellschaft, Berlin.
8. Brown Boveri, Baden.

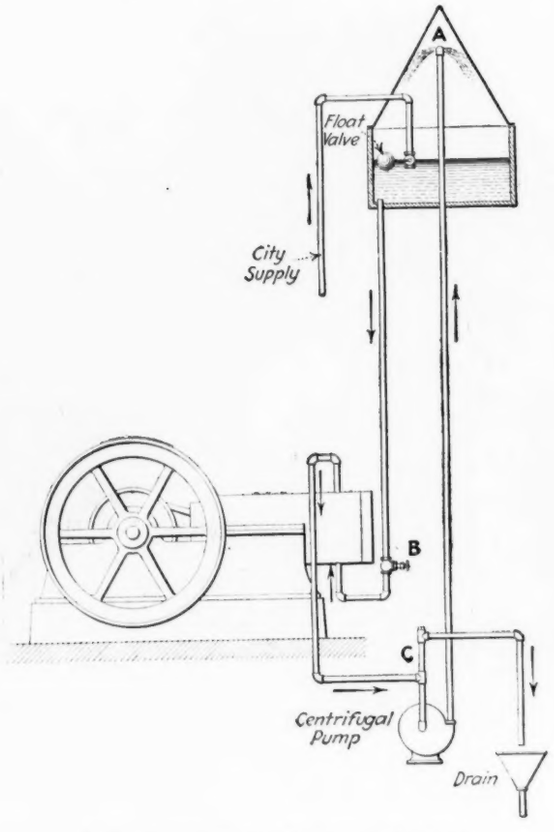


EXAMPLES OF STEAM TURBINES OF THE COMPOSITE TYPE

# Gas-Engine Cooling Water

By G. A. Field

The temperature of the jacket water should be controlled to suit the individual case. For instance, large engines require cooler cylinders than smaller ones, and oil engines often require slightly higher temperatures than those burning gas or gasoline, because of the tendency of the oil to condense on the cylinder walls. In automobile engines the jacket water often reaches the boiling point without any serious results. Under average conditions the temperature of the cooling medium on entering the cylinder jacket will not exceed 60 deg. F. and on leaving should not exceed 160 deg., 150 deg. being better practice. Should the temperature of the




Arrangement of Cooling System

cooling water exceed 185 deg., there is danger of deposit in the jacket.

As a safeguard against incrustation of the water jacket soda may be introduced, about a pound a month being used for every 17 cu.ft. of reservoir capacity. The jacket should be flushed out frequently. Another method is to fill the jacket space with one part sulphuric acid and ten parts water and allow it to remain over night.

For small engines, not hopper cooled, but using the thermo-siphon system, ordinary tanks or reservoirs are used. In this case the bottom of the reservoir should not be below the water outlet of the cylinder jacket, as the circulation is maintained solely by the difference in the density of the water due to the difference in temperature at the inlet and the outlet. The height of the water in the reservoir should not be less than four inches above the discharge of the return pipe. The capacities of reservoirs of this type are, of course, dependent upon the size of the engine and should be from 50 to 60 gal. per hp.

Water injection in various quantities directly into the combustion chamber is also used, but has not as yet displaced the water jacket to any great extent. A very small quantity of water is commonly injected with the fuel of oil engines to prevent preignition and also, by lowering the temperature of the burning charge, to prevent the decomposition of the fine particles of oil before evaporation is fully attained. This results in a decrease in the amount of carbon deposited in the combustion space.

It is estimated that about one-third the total heat supplied to the engine is carried away in the cooling water. Manufacturers guarantee a heat consumption of from 11,000 B.t.u. per b.hp.-hr. for full load, to 19,000 or 20,000 B.t.u. for one-quarter load. Therefore, with a safe allowance for ordinary working conditions of 12,000 B.t.u. per b.hp.-hr., based upon full-load rating, it will be seen that about 4000 B.t.u. per b.hp.-hr. must be carried away by the cooling medium.

As one B.t.u. is required to raise 1 lb. of water 1 deg. F., for an average range of 90 deg. it will require $4000 \div 90 = 44.5$ lb. of water, which is equivalent to approximately 5.35 gal. per b.hp.-hr.

Practical experience shows that small hopper-cooled engines require from 0.3 to 0.6 gal. of cooling water per b.hp.-hr., while larger engines using forced circulation should have a pump capacity of from 10 to 15 gal. per b.hp.-hr.

An economical and efficient arrangement for recooling the jacket water, and also providing for loss due to evaporation or danger of the pump failing to operate, is shown in the attached sketch. The cooling tower may be placed on the roof at some elevation above the engine, and it should be in the open air. The hot water is delivered along the under side of the ridge at *A* and is sprayed outward on each side against the sloping sides through holes drilled in the pipe. The sides are made of fine woven wire, which permits free passage of the air and carries the water down to the open tank below. Water enters the tank from the city supply pipe until the level reaches a sufficient height to close the float valve.

After starting the engine, the valve *B* is opened and the centrifugal pump is started. The pump, being at the lowest point in the line, is always primed and the cooling water immediately takes the circuit downward from the tank through the water jacket of the engine, into the pump, and upward to the spray and back to the tank.

Should the pump fail to operate properly or stop altogether, the circulation will still be maintained; the water being constantly replenished by the city-supply pipe through the automatic float valve. After passing through the cylinder jacket the water, instead of passing through the pump, will flow upward through the pipe *C* and into the drain. By means of the float valve, all losses due to evaporation are replenished, making the system automatic.

The pump should, if possible, be belted to the engine, but may be run by an electric motor. In shutting down the engine the valve *B* is, of course, closed and the pump stopped.

**Manual Training Schools** in the United States numbered 279 in 1913, with 65,699 students.

## A Large Wooden Flywheel

Engine wheels of large size, especially those designed for high rim speed, are, in many instances, being built up with wooden rims, and many old-style cast-iron wheels on large engines are being replaced with the modern wood-rimmed wheel. Such a change has recently been made at the large cotton mill operated by the Berkeley Co., at Berkeley, R. I. The two independent Corliss engines at this plant, each having a separate flywheel and shaft, are operated together as a cross-compound, in conjunction with an independent condensing apparatus. The cylinder of one is 26 and of the other 52 in. in diameter, each having a 72-in. stroke. The wheel of the high-pressure engine, 25 ft. in diameter and 88 in. width of face, was originally of cast iron, the centers being in two pieces forced upon the shaft, and the arms and segments, eight of each, being cast separately.

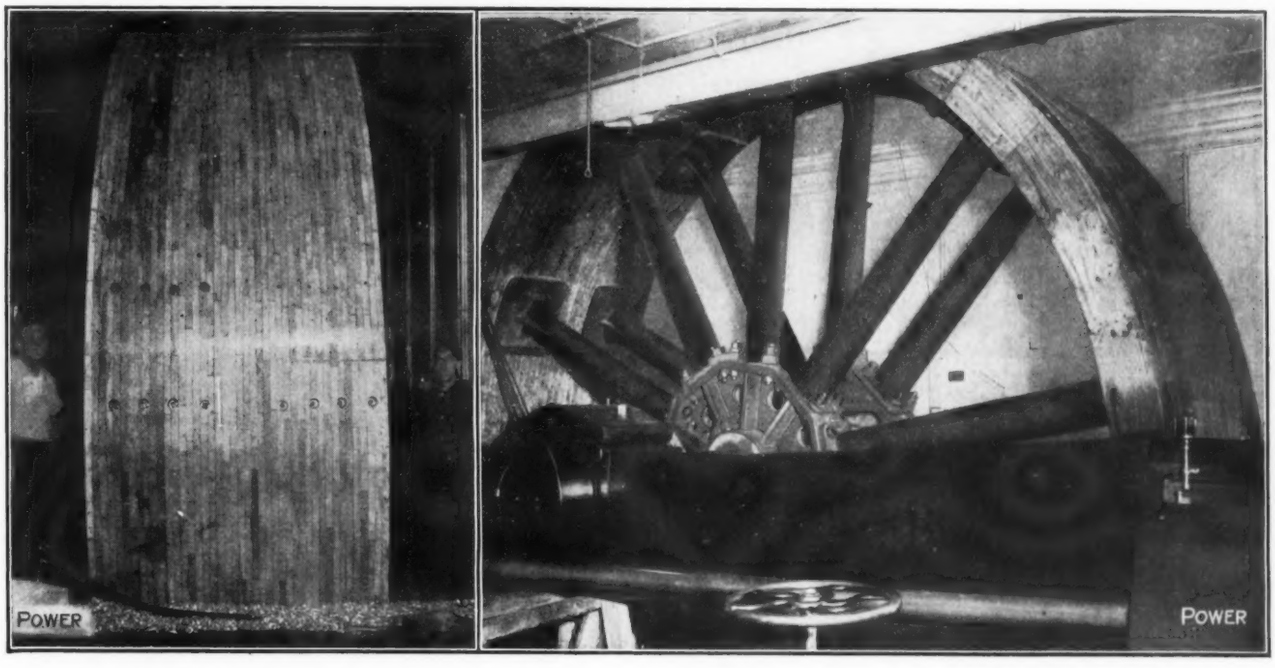



FIG. 1. TWENTY-FIVE-FOOT FLYWHEEL WITH WOODEN RIM AT BERKELEY (R. I.) COTTON MILLS

This wheel was designed for about 50 r.p.m., and at the time of its installation so large a wheel, with only eight arms and eight rim segments, was considered to have an ample factor of safety when running, as this ran, with a rim speed of 65.45 ft. per sec. Later, the speed of the engine was increased to 60 r.p.m., giving it a rim speed of 78.54 ft. per sec., and as a matter of insurance, the built-up iron wheel was stayed for additional strength by applying wrought-iron bands on the shaft on each side of the wheel hub and having steel stays reach therefrom to the bolts securing the joints of the rim segments; these stays being 16 in number, twice as many as there were arms in the wheel.

In this manner the wheel was operated for a considerable time and was carefully inspected at given periods. Some months ago, during such a week-end inspection, cracks were noticed in the pad of one rim segment at the point where the wheel arm was bolted to the rim, and while the cracks found were not such as would indicate any necessity for condemning the wheel, it was thought wise to apply special wrought-steel patches to strengthen the pad on the rim segment, to prevent further development of the cracks. Such repairs as could be made between a Saturday night and the following Monday morning were attended to, and the wheel was operated with the mill load the greater part of the following Monday morning, when it was thought wise to shut down and note conditions. It was found that the pad at both ends of the segment adjoining the one originally cracked had developed fractures, and special steel patches had to be made up and applied to those segments before the engine could be again operated. Meanwhile, it had been determined that the wheel, cracked as described, was not safe to run, no matter how well repaired and patched up, and an order was placed for a modern wheel of such design as would permit the engine to be operated at any desired speed and still have a greater factor of safety than could be had with an all cast-iron wheel.

The new wheel was designed, built and installed by the William A. Harris Steam Engine Co., of Providence, R. I., and is of the same size as the one replaced, but the hubs are in two pieces, designed for a clamping fit to the shaft instead of being forced into place, and the arms, which are cast separately, are 20 in number, of oval cross-section and hollow, there being two sets of these arms, 10 in each set. The rim is 14 in. thick, built up of white pine, and while the new wheel is superior in every particular to the old type of wheel and has a greater factor of safety, it weighs only about the same as the iron one which it replaces, and does not, therefore, bring any greater weight on the main bearing, the original shaft being reemployed. The total weight of the new wheel is about 80,000 lb., of which 60,000 are in the iron work of the hubs and arms, and 20,000 in the wooden rim. The patched-up and repaired iron wheel was operated until the new one could be made ready, and then, during one week's shutdown, the old wheel was removed and the new one installed without removing the main shaft from the engine.

It will be noticed from the accompanying illustration that the new wheel center, while being in only two pieces, has double faces to accept two sets of arms, as described. This center complete weighs not far from 10 tons in itself, while the 20 arms weigh about a ton apiece.

The work of removing the old wheel and installing the new one was carried on under pressure, so to speak, and in cramped quarters, and the accomplishment of the complete job in one week's time was considered quite remarkable. The wood rim was built onto the arms, piece by piece, at the mill, each piece being securely spiked and

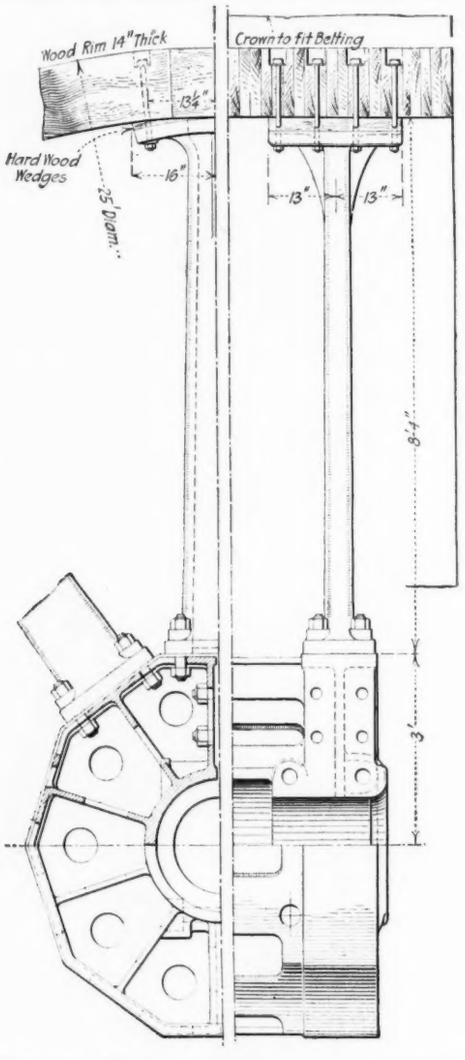



FIG. 2. SHOWING CONSTRUCTION OF LARGE WOODEN FLYWHEEL

glued, and then the completed rim was turned up true and faced off with three independent crowns for as many belts of unequal width.

The builders of this wheel have installed a number of such wheels at various places throughout the country, and make a specialty of their design and construction. They have recently converted two 15-ft. by 25-in. iron wheels into one wheel of 16½ ft. diameter by 54 in. face, by building a wood rim onto the faces of the old iron wheels, giving them greater strength, permitting higher rotative speed than that for which they were designed, and allowing the operation of one belt where two had previously been employed, while the cost of converting the old wheels into one was much less than would have been the cost of a new iron wheel to accomplish the desired purpose, and the time required was really not so great as would have been needed if the old wheels had been removed and a cast-iron one substituted.

## Just for Fun

*[More stories of stupidity and ignorance competing with "Some Original Ideas," as printed Jan. 19, 1915.]*

In one of the mills of this city there are two 72-in. by 16-ft. return-tubular boilers carrying 110 lb. steam pressure. The average load is about 176 i.hp. The boilers are kept reasonably clean and oil is used for fuel. One of the mill owners thinks he needs more capacity for steam generation because the engineer reports that the shell of the boiler becomes almost white-hot. After turning off the oil it gradually cools down to a cherry-red. —*A. L. Harris, McPherson, Kan.*

A few years ago I read in an article, "To stop knocking give your valve more lead." Very simple, thought I, although it had been my belief that the knock was in the piston and not in the valve. Besides, I had always thought that babbitt metal was better than lead. However, being a believer in following directions to the last letter, I took the valve, valve rod, and eccentric off, melted out the babbitt and substituted lead. I was firm in my conviction that it would be impossible to give the valve more lead than that. The knock did not stop at all; the article, therefore, was wrong. Nevertheless, lead is necessary and it does stop knocking.—*W. F. Schaphorst, New York City.*

A prize "bone-head" trick was pulled off on a new Corliss engine built by one of our largest concerns and erected by one of their shop men, but evidently changed afterward. At any rate complaint was made that the engine would not carry the load, in fact would "lay down" with about quarter load. One of the best shop men was rushed to the scene, arriving just before noon. On entering the engine room the very first thing he noticed was that the eccentric connecting-rod was connected to the upper pin on the rocker and the reach-rod connected to the center pin—in other words the two rods were reversed in position. The motion being so reduced, the valves were not given sufficient opening, hence the lack of power.

Taking in the situation at a glance, the shop man suggested going to dinner before beginning the task of putting the engine right, to which the engineer agreed. They started out together, but the shop man suddenly remembered having left his gloves, told the engineer to go ahead and order the dinner and he would go back after them. While in the engine room alone, he quickly reversed the connections, then hastened to join the engineer in a good dinner and "trimmings." On their return to the engine room he suggested that the engine be started up so that he might judge of its ailment. It, of course, behaved all right; then he suggested putting on the load, and to the wonder of the engineer it carried it with ease. The engineer was kept guessing to the full satisfaction of the shop man before being told what had happened.

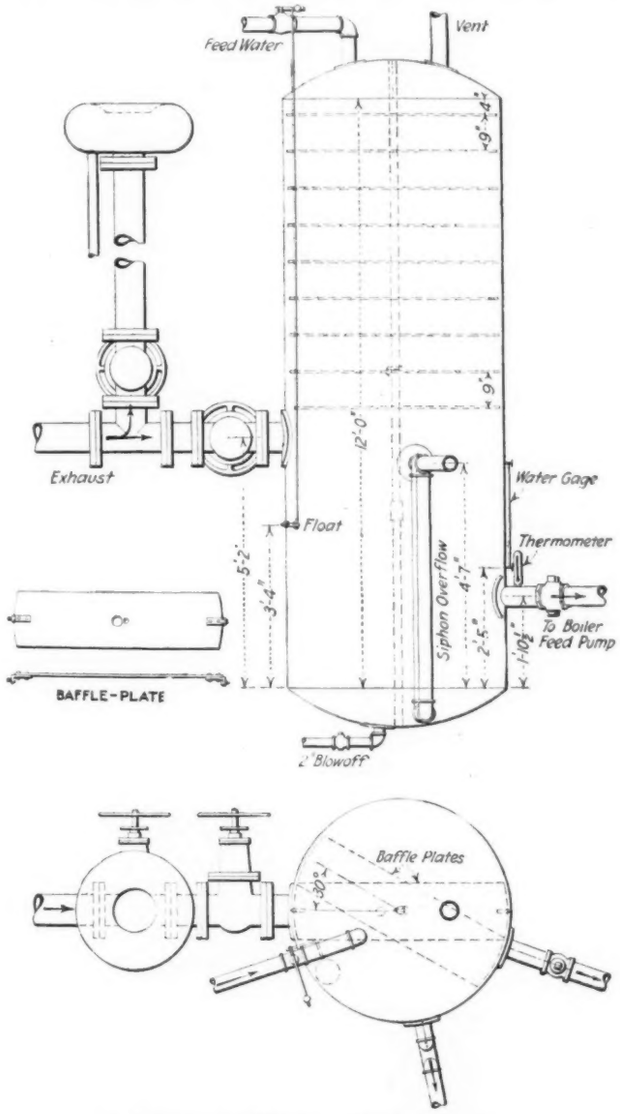
This is a "sure enough," "honest injun" true story.—*F. R. Compton. New York City.*

# An Open Feed-Water Heater

By F. F. Jorgensen

The illustration is of a home-made open feed-water heater used at a coal-washing plant. It heats all of the feed water for four 150 (rated) horsepower boilers. Exhaust steam is taken from a 13x16-in. double engine, a 12x14-in. single engine, a 13x12-in. single engine, the boiler-feed pump and a small coal-drag engine.

Nearly 9000 lb. of exhaust steam per hour is available. The back pressure is but little above atmospheric,




Details of Heater Construction

so that if all the steam were condensed 970.4 B.t.u. per lb. of steam would be obtained, or 8,733,600 B.t.u. in all.

About 15,000 lb. of feed water per hour is required and its temperature on entering the heater is very little above 32 deg. during the cold weather. The temperature leaving the heater remains practically constant at 212 deg. In raising the temperature of the water from 32 to 212 deg., 180 B.t.u. per lb. is required.

The heater was made from a discarded compressed-air receiver and it has been in service for a year, giving excellent results.

# Variation in Coal Constituents for Similar B.t.u. Values

By Henry D. Jackson

The recent and persistent agitation of the subject of purchasing coal on the B.t.u. basis is of value to both the buyer and the seller, but if coal were bought on the B.t.u. basis alone, it would be likely to prove an expensive method to the purchaser, for all coals having a high B.t.u. content are not necessarily good for use under the boilers of the plant for which the coal is being purchased.

Coal varies widely in ash, moisture, fixed carbon and volatile matter. The ash is, of course, a waste, and a low ash content is advisable because it means that less coal has to be fired and less waste material handled for a given evaporation. Moisture is a detriment because it must be evaporated and the heat to do this must come from the coal, and therefore, considerable moisture means a large waste of heat. Volatile matter may or may not be of value, depending upon what it consists of, as well as whether it can be burned to advantage under the boiler.

The table shows the range of constituents found in coals having the same B.t.u. value. Take the two top rows

TABLE SHOWING WIDE RANGE IN COAL CONSTITUENTS FOR APPROXIMATELY THE SAME B.T.U. VALUES

| Value in B.t.u. | Moisture | Volatile Matter | Fixed Carbon | Ash |
|---|---|---|---|---|
| 10,001 | 8.93 | 36.53 | 33.76 | 20.78 |
| 10,264 | 3.53 | 20.75 | 47.85 | 27.87 |
| 10,816 | 10.83 | 36.24 | 39.75 | 13.18 |
| 11,142 | 9.04 | 29.65 | 45.57 | 15.74 |
| 12,195 | 3.12 | 35.75 | 47.67 | 13.96 |
| 11,906 | 5.59 | 25.05 | 53.28 | 16.08 |
| 12,958 | 4.49 | 40.55 | 47.43 | 7.53 |
| 13,129 | 3.24 | 17.46 | 66.69 | 12.61 |
| 14,107 | 1.75 | 36.77 | 55.14 | 6.34 |
| 14,024 | 2.14 | 16.83 | 71.91 | 9.12 |

of figures, for instance; the moisture varies between 8.93 and 3.53 per cent., the volatile matter between 36.53 and 20.75; fixed carbon between 33.76 and 47.85, and the ash between 20.79 and 27.87, yet the B.t.u. content is practically the same.

For most boilers as set at the present time, the second coal shown on this list would probably be more satisfactory as regards evaporation than the first, although its ash content is higher. This is due to the higher fixed carbon and lower moisture. Following these figures right through, one can readily see that there is a wide variation in coals. Therefore, the buyer should take into account more than the B.t.u., and in order to be sure that the coal purchased is the best for the purpose, it is advisable to burn it under the boilers and to determine the evaporation, and then buy by specification that coal which gives the best evaporation for the lowest price. Test every car of coal and see that it comes up to specification. Put a bonus and forfeiture clause in the specification, which will give a bonus for an increase in the fixed carbon or decrease in the volatile, ash or moisture, and exact a forfeiture or a decrease in price for excess of ash, moisture, or volatile matter or decrease in fixed carbon, the object being to obtain a coal having a maximum of B.t.u. with a minimum of waste material. The specification should limit the minimum amount of fixed carbon as well as the maximum amount of ash and moisture acceptable.

**A New Organization** called the Institute of Industry and Commerce has been formed in London. The institute hopes, among other things, to stimulate and encourage standardization in methods of production, organization and distribution, to bring about closer coöperation between science and management and labor in industry, and to consider all legislation which may affect industry.—Foreign Exchange.

# Test of Turbo Air Pump

*SYNOPSIS—A pump which apparently produces over 100 per cent. vacuum, the absolute pressure in the chamber being less than that of water vapor present at the observed temperature.*

The editor was privileged to see recently, at the works of the Wheeler Condenser & Engineering Co. at Carteret, N. J., a turbo air pump under test. This pump is of the type in which the air is expelled by a succession of rapidly moving water pistons. Fig. 1 shows the action diagrammatically, the rotating impeller throwing off streams of water at a high velocity, which are broken up by the pointed divisions of the compression channels into plugs which continue their course through the channels, pushing the air and noncondensable vapors before them. The pump in this form was developed by the Allgemeine Elektricitäts Gesellschaft, the General Electric Co. of Germany, and is built by the Wheeler Condenser & Engineering Co. Fig. 2 shows the exterior of the impeller and the renewable entry piece at the commencement of the compression channels, and Fig. 3 the pump in section.

The pump under test had a capacity suitable for about a 20,000-kw. condenser. The purpose was to determine how nearly it would come to maintaining a theoretical vacuum when different amounts of air were allowed to enter. The pump was arranged for testing, as shown in Fig. 4. The plugs at *AAA* controlled carefully calibrated openings through which known amounts of air were admitted. The temperature of the incoming and outgoing hurling water was measured by the thermometers at *B* and *C*, respectively. The mercury column at *D* indicated the vacuum, and the V-notch weir *E* enabled the amount of hurling water used to be determined.

The temperature of the hurling water as measured by the thermometer at *B* was 93 deg. F., and one would naturally say that the absolute pressure in the chamber *F* could not be less than 1.556 in., the pressure of water vapor at 93 deg. As a fact, however, the mercury gage *D* connected with this chamber showed a vacuum of 28.8 in.

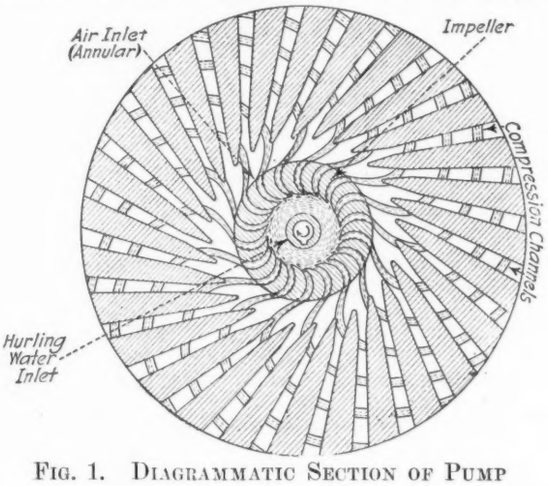



FIG. 1. DIAGRAMMATIC SECTION OF PUMP

The barometer at the time read 30.02 in. If these were right the absolute pressure in the chamber was 30.02 — 28.80 = 1.22 in., or only 0.784 of the tension of water vapor at 93 deg. with which it was supposed to be in con-

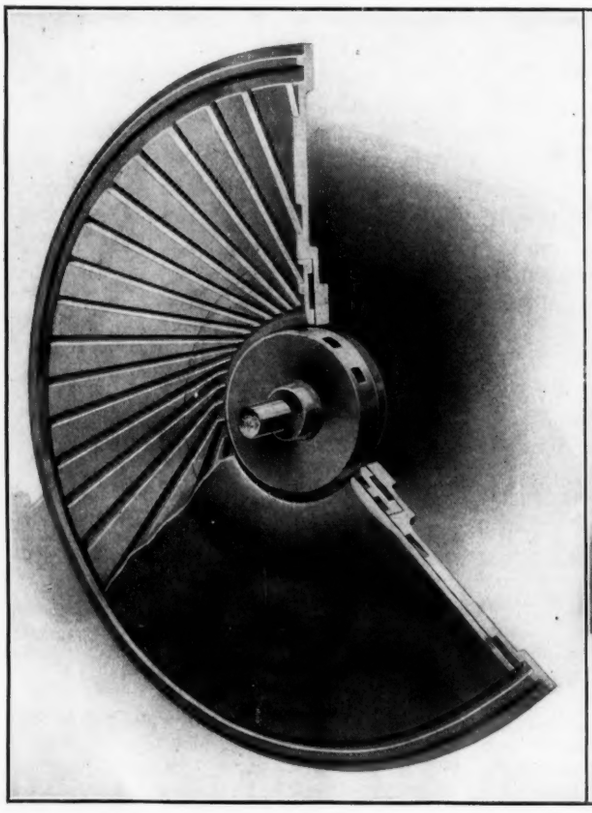


FIG. 2. IMPELLER AND COMPRESSION CHANNELS

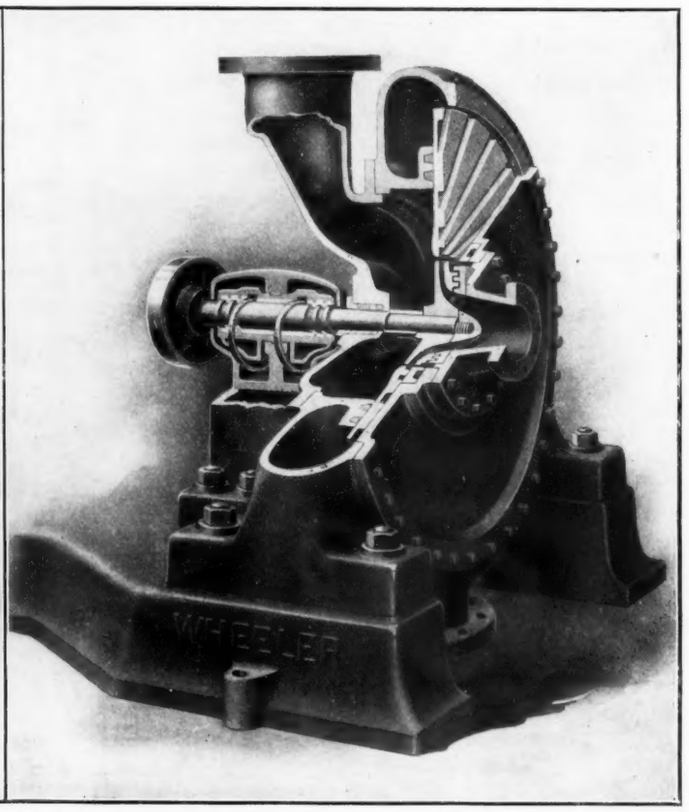



FIG. 3. SHOWING IMPELLER IN SECTION AND IN SITU

tact. It was not until just after the fourth observation, as shown in the fifth column of Table I, when the air was being admitted at the rate of 32 cu.ft. per min., that the pressure in the air inlet became greater than that due to

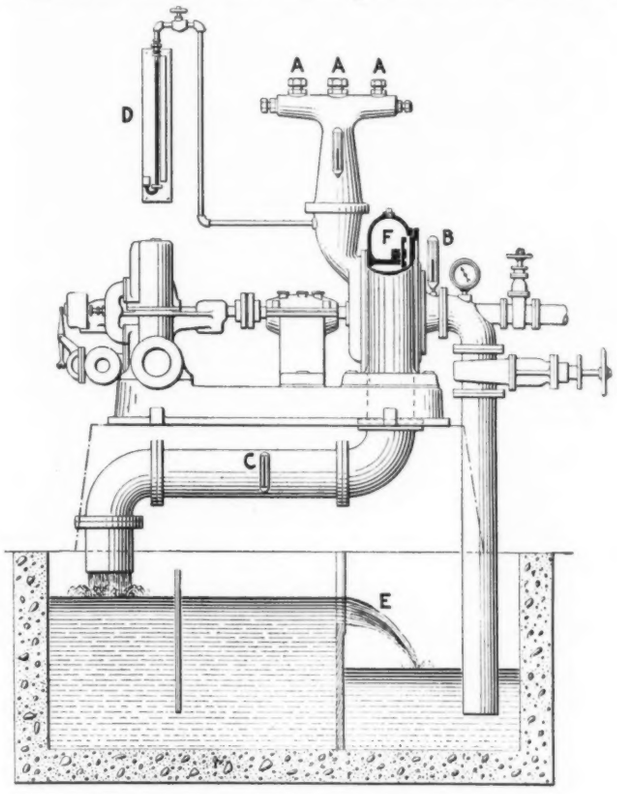



FIG. 4. PUMP ARRANGED FOR TESTING

the temperature of the hurling water. An explanation suggested by the engineers of the condenser company is, that as the water is subjected to the action of the vacuum a small amount evaporates under the reduced pressure and a cooling effect occurs, resulting in a lower temperature and vapor tension, or higher vacuum, in the immediate vicinity of the surface of the water.

It is customary at the Wheeler works to refer all vacuums to a 30-in. barometer. This is done in the sixth column of the table, and the ratio of the vacuums so found to the vacuum (28.444 in.) corresponding to the tension of aqueous vapor at 93 deg. and a 30-in. barometer is given in the eighth column. This ratio would, however, be different for each barometric pressure taken as a base.

TABLE I

| Barometer Reading In. Mer. | | Observed Vacuum In. Mer. | | Absolute Pressure at Air Inlet In. Mer. | | Pressure of Aqueous Vapor at 93 Deg. F. In. Mer. | | Ratio of Condenser Pressure to Vapor Pressure at 93 Deg. F. | Observed Vacuum Referred to 30-In. Bar. | | Vacuum Corresponding to Pressure of Aqueous Vapor at 93 Deg. F. Referred to 30-In. Bar. | | Ratio Observed Vacuum to Vacuum Corresponding to 93 Deg. F. Both Referred to 30-In. Bar. | Free Air, Cu.Ft. per Min. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 30.02 | — | 28.80 | = | 1.22 | ÷ | 1.556 | = | 0.784 | 28.78 | ÷ | 28.444 | = | 1.0118 | 0.0 |
| 30.02 | — | 28.77 | = | 1.25 | ÷ | 1.556 | = | 0.803 | 28.75 | ÷ | 28.444 | = | 1.0108 | 3.0 |
| 30.02 | — | 28.73 | = | 1.29 | ÷ | 1.556 | = | 0.829 | 28.71 | ÷ | 28.444 | = | 1.0094 | 8.7 |
| 30.02 | — | 28.60 | = | 1.42 | ÷ | 1.556 | = | 0.913 | 28.58 | ÷ | 28.444 | = | 1.0049 | 14.0 |
| 30.02 | — | 27.86 | = | 2.16 | ÷ | 1.556 | = | 1.388 | 27.84 | ÷ | 28.444 | = | 0.9788 | 32.0 |
| 30.02 | — | 27.10 | = | 2.92 | ÷ | 1.556 | = | 1.877 | 27.08 | ÷ | 28.444 | = | 0.9520 | 46.0 |
| 30.02 | — | 26.60 | = | 3.42 | ÷ | 1.556 | = | 2.198 | 26.58 | ÷ | 28.444 | = | 0.9345 | 57.0 |
| 30.02 | — | 24.90 | = | 5.12 | ÷ | 1.556 | = | 3.290 | 24.88 | ÷ | 28.444 | = | 0.8747 | 89.0 |
| 30.02 | — | 24.05 | = | 5.97 | ÷ | 1.556 | = | 3.837 | 24.03 | ÷ | 28.444 | = | 0.8448 | 108.0 |
| 30.02 | — | 21.75 | = | 8.27 | ÷ | 1.556 | = | 5.315 | 21.73 | ÷ | 28.444 | = | 0.7640 | 165.0 |
| 30.02 | — | 20.00 | = | 10.02 | ÷ | 1.556 | = | 6.440 | 19.98 | ÷ | 28.444 | = | 0.6024 | 222.7 |

In Table II are given the results of a test of a pump of the same type, but of relatively small size, connected to a surface condenser and with cooler water, the condenser working at about half of its rated capacity. Notice the close agreement between the exhaust-steam temperature corresponding to the vacuum (Col. 3) and as observed (Col. 4).

## Steam Piping

While the work of planning and erecting steam piping seems of minor importance, its proper execution affects the economy and continuity of the plant's operation to a greater degree than an equivalent expenditure on any other feature of the average power plant. The factor of safety usually employed for pipes and fittings is from six to ten times the working pressure they are intended for, yet constant annoyance and even disaster follow their improper erection or later neglect. Among the chief causes of failures of pipe systems are water-hammer, expansion and contraction, which produce distortion, vibration, misalignment, poor workmanship at joints, and corrosion.

Water-hammer may be produced by a great variety of

TABLE II

| Time | Vacuum Referred to a 30-In. Bar | Temp. Corresponding | Temp. of Exhaust by Thermometer | Temp. Hurling Water In | Temp. Hurling Water Out | Temp. Circulating Water In | Temp. Circulating Water Out | Temp. Condensate |
|---|---|---|---|---|---|---|---|---|
| 12:30 | 29.64 | 50 | 50 | 37 | 39 | 37 | 44 | 46 |
| 1: | 29.64 | 50 | 52 | 38 | 40 | 38 | 43 | 47 |
| 1:30 | 29.59 | 53.5 | 54 | 38 | 40 | 38 | 44 | 48 |
| 2: | 29.59 | 53.5 | 54 | 38 | 40 | 38 | 44 | 48 |
| 2:30 | 29.59 | 53.5 | 54 | 38 | 40 | 38 | 44 | 48 |
| 3: | 29.54 | 56.5 | 58 | 37 | 40 | 37 | 46 | 46 |
| 3:30 | 29.79 | 53.5 | 54 | 37 | 40 | 37 | 46 | 47 |
| 4: | 29.64 | 50 | 52 | 36 | 38 | 36 | 44 | 45 |

conditions, some of which are difficult to forestall or identify, but the effect is usually the same—broken fittings or at least loosened joints, resulting in more or less serious damage. Any depression, or pocket, below the drainage level of the pipe line may, under certain conditions, accumulate condensate or water enough to produce water-hammer, therefore such pockets should be avoided. A valve so placed that water may accumulate above it when closed, constitutes one of the most frequent sources of water-hammer. When such a location is unavoidable a drain should be tapped in immediately above the valve, with the small drain valve easily accessible to the operator when at the main valve. A short open-end drain pipe is usually best, so that the operator may be sure the drain is not obstructed and that the steam is reasonably free from moisture, before opening the main valve. This is intended to be independent of and in addition to the regular drainage system. Reducing fittings on horizontal runs are frequently responsible for water pockets. Owing to the rapid rate of condensation, water-hammer may occur if steam is too rapidly turned on long lines when cold, even though sufficiently graded and drained for ordinary duty. It should be borne in mind that disastrous water-hammer may occur at low pressure—even below that of the atmosphere.

Expansion and contraction, when no adequate allowance has been made to relieve the strain so produced, always tend to loosen joints and often distort valves and cause fittings to break. Allowance should be made for elongation with an increase of temperature to which the pipe is subjected, equal to the difference in degrees multiplied

by the length in inches and divided by the constant 150,000, or 0.6 in. for each 100 deg. per 100 ft. of pipe. The most commonly used form of expansion absorber is the ordinary, long-radius bent pipe. So called swing joints are also used, consisting of screw fittings so placed that expansion or contraction will cause them to change their relative positions by turning on the thread. Two pairs of elbows connected by short lengths of pipe are usually necessary. The threads for such work should be well formed and ground in with emery and oil to make as perfect a ground joint as possible, and after wiping off the emery they should be well lubricated and not made up too tightly. The extra resistance to the flow of steam caused by the elbows is an objectionable feature in addition to the difficulty of maintaining a steam-tight joint. The slip joint, consisting of a sleeve, smooth and true, attached to one section of the pipe and sliding in a stuffing-box attached to the other section, is quite commonly used and has advantages and uses peculiar to itself. Another form is made of a pair of large circular flexible diaphragms of steel, copper or other metal bolted to the flanges of the two sections of pipe and bolted together at their periphery with a flange of large diameter between. This forms an expansion joint of considerable merit. A corrugated copper sleeve, with supporting rings and having a flange attached at each end, is used where the expansion is not too great. Care should be taken not to tax this joint beyond the limits recommended by the manufacturer. Ball-and-socket joints and sometimes unions are utilized to avoid excessive expansion strains.

Vibration is usually caused by motion of the engine or other machinery to which the pipe is attached and the pulsating of the steam in the pipe. If this motion, or vibration, is rapid and violent the life of the pipe line will be uncertain. In cases where there are several high-speed engines which cause the pipes to vibrate in unison part of the time and out of step at other times, the strain becomes particularly severe at intervals. Lines should be free to expand lengthwise, but their motion in other directions should be carefully limited.

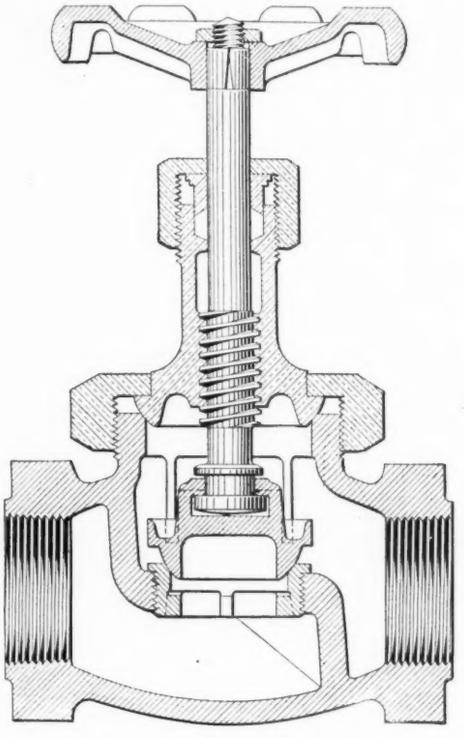
Misalignment puts severe strain on the pipe and fittings, either in connection with expansion and contraction or independently. If the pipe line is not straight, because of crooked threads or flanges, the difficulty will usually increase under heat changes, although joints that have to be forced out of their natural positions or strained when cold may sometimes take an easier position when heated; but such strain should be carefully avoided.

Poor workmanship will soon manifest itself when the pipe system is subjected to the foregoing conditions in addition to the pressure sustained. Work poorly done through carelessness or ignorance is scarcely less criminal than if so done with intent to do injury. Ignorance is no excuse and carelessness is less than none—it is a confession.

External corrosion is sometimes serious when a combination of heat and moisture is augmented by a trace of acid or other corrosive element in the pipe covering. This condition is more frequently met with in underground construction where, incidentally, it is the more serious on account of the inaccessibility of the line. It can usually be avoided by a protective coating applied to the pipe before the insulating covering is put on. Internal corrosion is more common in boiler-feed lines and return lines from steam-heating systems and is sometimes due to the action of warm distilled or nearly chemically pure water, which readily attacks wrought iron or steel. Water not over 80 per cent. pure is not likely to attack the piping seriously, but if air is admitted with warm water containing a trace of acid there is likely to be corrosion. Cast-iron or brass pipe will usually withstand indefinitely the action of water fit for boiler feed.

## Powell "Irenew" Valve

The Powell "Irenew" valve, a sectional view of which is shown, has recently been developed by the William Powell Co., Cincinnati, Ohio. The main feature is the removable horseshoe disk, which is arranged to slide over the head of the stem into a socket, thus permitting it to swivel freely. When it is desired to remove the disk from the stem for regrinding or renewing, it is merely slipped from the socket. Should it become needful to regrind the valve, it is not necessary to disconnect it from the pipe, but by simply releasing the bonnet and unscrewing the large hexagonal nut the valve bonnet may be withdrawn and a pin fitted through a hole in the valve-stem head to lock the disk.



"Irenew" Valve in Section

The removable disk is made of a noncorrosive metal applicable to most temperatures of superheated steam. To remove the seat, a screwdriver or other flat tool is used, which engages with the lug projecting from the inner circle; the seat is screwed out of the valve and a new one inserted.

**Municipal Plant Lowers Rate**—The City Council of Two Harbors, Minn., has ordered the rate charged by the municipal lighting plant reduced to 6c. per kw.-hr. for lighting instead of 8c., as heretofore, and a new rate of 3c. per kw.-hr. for power. Consumers who take the power rate will be obliged to install a separate meter. The municipal power plant is a paying proposition, and the city officials feel that they can afford to make the reduction in rates.

# Editorials

## Depreciation as a Practical Problem

To many engineers the word depreciation visualizes difficult and wearisome arguments upon the present and past values of power-plant equipment by theorists before commissions passing upon the right of corporations to issue more stock, maintain existing prices for service, or consolidate with others of their kind. To others depreciation means the underlying reason why the company had to buy a new condenser or feed pump last week, after getting a good many years of satisfactory service out of the equipment. To still others it means the cause of getting eight hundred dollars for a compound engine in perfectly workable condition, which cost originally six thousand dollars, but which has been obliged to make way for a turbo-unit that will generate more horsepower-hours per cubic foot of space in a day than the faithful old cross-compound could produce in a fortnight.

In technical literature there is hardly a subject on which more purely speculative matter has been written than on depreciation. Hundreds of pages in court and commission cases have been devoted to mere definitions of the word, and assumptions by the thousands have been made and will be made as to values and percentages to be allowed for depreciation in one form or another, in trying to prove some administrative policy just or some existing set of charges proper. Probably depreciation will never be reduced to a plain "two-and-two-make-four" basis while progress continues in equipment and plant design, but men closely associated with such plant and equipment can do much toward ridding the subject of some of its most glaring uncertainties if they attack the problem systematically and are accorded the coöperation of their employers.

Depreciation is, finally, a matter of plant life. How it shall be offset is a problem for the statisticians, but the basis upon which the men behind the gratings and wired-glass windows are to proceed has a direct and inescapable origin in the work of the engineer. How little we really know about this life question and how much we assume! Surely, it is time for the engineer to begin to contribute his observations and judgment in a broader way upon this important problem, which is always the unknown coefficient of the manager's equation of probable cost. Where can the engineer lay hold of the matter effectively enough to help his employer, to add some specific solvent to the mass of undigested and hypothetical data which is congesting the modern industrial library?

There is only one way to go about this task and that is by keeping a record of the installation, repairs, replacements, failures, and final disposition of every piece of apparatus affecting an estimate of the probable life of similar equipment at any future time; of studying the influence of performance and of idleness upon wear and tear and upon fitness for the service of today and of day after tomorrow; and by maintaining sufficiently complete records to enable the owner of the apparatus at practically any time to determine with least delay the total outlay of money upon it to date compared with the initial cost. It is not too late to begin to collect such material in many plants where the original equipment still handles the daily service. Every time a piece of machinery goes into commission fresh from the factory, the operating engineer should be permitted to acquaint himself, if he will, with its initial cost in as much detail as necessary; and the dates on which spare parts are substituted, with the cost of so doing, should be set down as part of a definite record which will enable the expert ultimately to judge the probable life of such equipment without guessing.

True, the mere cost of repairs and spare parts inserted may not throw light on the life of a machine as a whole, but here is exactly where definite data are useful in marking off life zones which will at least indicate where money must be spent to make good the aging of equipment and where enough durability can be anticipated to reduce the annual sinking-fund allowances for final replacement. There has been too much temptation in the past to assume all-around depreciation rates on aggregations of apparatus having enormous differences in life—a policy justified by the need of doing something constructive to establish a fund capable ultimately of putting in the equivalent in capacity of the depreciated equipment, but none the less a policy which must sooner or later give way to the more scientific plan of basing life and total cost estimates on data gathered in plant and field. There is room for real research in this department of engineering economy.

## Placing the Blame

If there is complaint that water powers in the public domain in the West are being withheld from use, the blame must rest with the men and interests responsible for the legislative methods which made impossible the passage of the Adamson and the Ferris bills, urged by the Secretary of the Interior, approved by the President, and indorsed by the leading conservationists of the country. Under cover of specious arguments for states' rights, filibusters on buffer bills, and senatorial courtesy, the same senators and the same interests which defeated water-power legislation seven years ago have again made it impossible this year.

The remaining water-power sites in the public domain are of enormous value, controlling as they do the key to the development and use of the vast water powers in Western cañons and streams. Many other valuable power sites once owned by the nation have been acquired by speculative and monopolistic private interests in the past for little or no return to the Government, and have been capitalized at large values on which power users have been required to pay interest in the form of power rates. Other sites are being held unused by private owners who paid nothing, or next to nothing, for them, in anticipation of the needs of communities not yet developed, or for the purpose of maintaining rates for power supplied from plants already in operation. While independent interests

concerned in hydro-electric development have generally expressed willingness to accept the terms of the bills before the late Congress, some of the larger and politically more influential interests refused even to consider the terms.

President Roosevelt withdrew the power sites which continue in the ownership of the people to prevent them from being gobbled up by the water-power trust. The "interests" at that time were powerful enough in Congress to prevent enactment of legislation that would allow use of these withdrawn sites under government control and with limited terms of occupancy and use. These same interests have so far blocked and defeated the efforts of the present Administration to secure legislation of the same nature. It is apparent that their purpose is to tire out the Government, in the hope that eventually these valuable sites may be given away, as have others in the past. It so happens, however, that the nation is fairly well informed nowadays of the value of these remaining national assets, and there is little probability of public opinion ever agreeing to turn them over as a free gift to any individuals. Fortunately for the country, delay in water-power legislation means nothing worse than delay in development. It is a big stake for which the trust is playing, but it has a forlorn hope of winning, and the longer obstructionists block legislation that would make regulated development possible, the stronger the growing sentiment for public development and operation is likely to become.

## Efficiency Engineering

Many engineers are beginning to see a lucrative field in applying the principles of efficiency to engineering, particularly to factory power plants. Many have already gained considerable success, which has tempted others to follow their lead.

Efficiency applied to power plants means actually the elimination of wastes, these wastes being usually the result of faulty engineering when the plants were designed. The owner of an inefficient plant is often the victim of an inefficient or an unfit engineer. This is a reflection on both the owner and the profession.

No engineer worthy of the name is really anything else than an efficiency engineer; his whole course of study and practice is to adapt nature's law to practical needs economically. No standard textbook teaches us to design otherwise than economically. The laws relating to the transformation of heat energy have been known for years. The heating of feed water, economical sizes and covering of steam piping, the heating value of exhaust steam, etc., are not new. Certain refinements in the apparatus used in power work have been made, and special equipment has been devised, much of which enables improvements to be made, looking to the saving of both labor and coal. At the same time, many of the new appliances are got up to sell. The uninitiated owner frequently falls for the expert salesmanship employed to sell these devices. Some salesmen call themselves, and really believe they are, efficiency engineers, owing, no doubt, to the ease with which they are able to dispose of their goods. This practice has given the engineer considerable trouble in the work he is now undertaking, as owners and superintendents have become somewhat skeptical.

The engineer who often meets with scant tolerance at the hands of the owners also has himself to blame, outside of the bad effect caused by the exploiting of bad appliances. Looking at the matter a little deeper, we find that almost anybody can hang out his shingle and practice engineering. If plausible enough, he can and does secure work which afterward needs considerable attention and expense to make it perform economically. Then again, there is almost a department-store variety of competition among some engineers, in price cutting, resulting in the job being a repetition of this practice.

We are inclined to believe that these methods have called attention to the need for the efficiency engineer far more than the so called recent discovery of efficiency, or the availability of new apparatus to secure economy, the principles of which are old and well understood.

Other professions, such as the medical and legal, require their members to pass examinations and meet certain requirements before permission is given them to practice. The profession of engineering, although really the oldest of them all, puts no restrictions on its followers. Would it not be best for all concerned if consulting as well as operating engineers were prohibited from practicing without a license?

## Skepticism as an Asset

Orthodoxy has little place in science. Skepticism—the insistent desire to be "shown," unwillingness to take things for granted—is a real asset to the engineer. It makes him uncomfortable enough, as everyone knows, but it helps him master his profession and makes him of increasing value to his employer.

It is unsafe to assume that a thing cannot be done merely because someone says so. Reports that a certain policy or practice is impossible must be checked before they are accepted. Suppose, in a large plant, a subordinate engineer investigates heat losses in certain piping and reports that nothing further can be done to remedy the situation. If the chief accepts such a report without checking its reasoning and conclusions, is he much better than a rubber stamp?

Coöperation and dependence upon the work of others are absolutely necessary today in engineering as well as in commercial activities, but a certain class of problems needs to be handled by direct methods, with routine thrown out of the window. All jobs which look impossible are of this class. Plenty of them *are* impossible, financially or physically, but the point is not to make any assumptions. If it is a report to the "boss," let it carry the convincing facts and arguments, so that it can be checked at the first reading. If it is a job handed down to the engineer from his superiors, let it be analyzed from every possible viewpoint before the answer is sent back that it is not feasible to carry out the plan.

These are more than generalities. They fit into daily experience. They teach that merely glancing over a report or a drawing and putting on the "O.K." with one's initials without a real check of the work, is largely wasted energy, economically unjustified. Question every proposition; make it prove its right to live; and by so doing cultivate the true scientific spirit which, combined with a sense of financial proportion, keeps the engineer high among the intellectual and constructive workers of the world.

# Correspondence

## Relative Water Required by Refrigerating Systems

Referring to G. B.'s question, page 311, Mar. 2 issue, I should judge that he wished to know the difference in quantity of water required for cooling, regardless of the kind of prime mover used. If both systems were motor-driven the absorption system would require three times the amount required for a compression system of equal capacity.

The quantity of water needed in the ammonia condenser would be the same in either system. To overcome the heat of absorption we need about twice the amount required in the condenser, making about three times the quantity for the absorption system that is needed when the compression system is used. When the strong-liquor pump to the absorption system is steam-driven its exhaust and other exhaust steam available can be used in the ammonia still, the condensate from the still can be returned to the boilers, and makeup water for the boilers can be taken from the water that has passed the condenser in either system.

In small absorption systems where closed ammonia condensers are used, it is usual to pipe the discharge from the condenser through the cooling coils in the absorber. When this is done there will be a slight rise in the temperature of the water after passing the condenser, but there will be a rise of about 30 deg. F. after leaving the absorption cooling coil. I have used this water to advantage in supplying a hot-water boiler when there was a demand for hot water about the place.

C. E. Bascom.

Westfield, Mass.

## Boilers for Isolated Plants

The following criticism is directed at errors, as they appear to the writer, in C. L. Hubbard's article on "Boilers for Isolated Plants" in the Feb. 16 issue.

1. About the middle of the first column, page 233, we read: "The heat absorbed by the water in the boiler per pound of coal burned =

$$\frac{970.4 \times W \times q \times f}{w}$$

where

$W$ = Apparent weight of water evaporated in pounds per hour.
$q$ = Quality of the steam.
$f$ = Factor of evaporation.
$w$ = Weight of coal burned, in pounds per hour.

It is theoretically incorrect to introduce the quality of the steam into the computations at this point. It should have been taken into account in figuring the factor of evaporation.

The total heat of dry saturated steam is made up of two parts, the sensible heat of the liquid and the latent heat of vaporization. If we are dealing with wet steam of quality $q$, we have for its heat content all of the heat of the liquid plus $q$ per cent. of the latent heat of vaporization. Obviously, it is incorrect to take $q$ per cent. of the sum of both parts when we should have used one part in its entirety and $q$ per cent. of the other part.

2. Near the bottom of the same page, is stated: "W.R. = Water rate of the engine under given conditions of *feed-water temperature and* steam pressure . . . ." The steam consumption of the engine is independent of the feed-water temperature and the words in italics should be omitted.

3. The last sentence of the article, near the top of page 234 reads: "All heating requirements are reduced to pounds of steam per hour and the result divided by 34.5 to find the boiler horsepower." This should read: "the result multiplied by the factor of evaporation and divided by 34.5 to obtain the boiler horsepower."

T. B. Hyde.

Lakewood, Ohio.

## Calculating Pump Slippage

The letter on the subject of calculating pump slippage, by George L. Sullivan in the Dec. 29, 1914, issue, page 928, indicates a practical, though somewhat inaccurate way of determining pump slippage.

The slippage of a pump depends not only on the amount of the fluid slipping by the piston, but a great deal of it is due to the fluid running back before the valves close. Therefore, at medium speeds, the cylinder is not quite filled to its capacity at each stroke of the piston. Thus, it is evident that unless the slippage is determined by a calibrated flow meter, or by weighing the water, the result will be inaccurate.

For small and medium-sized pumps it is practical to connect a number of barrels or other receptacles in series, so that by connecting the discharge with one of the barrels, the water will run over and flow into the next barrel.

Knowing the capacity of the barrels, the slippage may be determined by subtracting the number of gallons of water in the barrels from the theoretical capacity of the pump, or,

$$S = \frac{aln}{231} - Q$$

where

$S$ = Pump slippage;
$Q$ = Quantity of water in the barrels in gallons;
$a$ = Area of the cylinder in square inches;
$l$ = Length of stroke in inches;
$n$ = Number of discharge or working strokes required to fill the barrels.

The pump should be allowed to make a few strokes before discharging into the barrels. Care should be taken to get the pump up to the normal working speed and pressure as soon as possible.

For large pumps this method is not practical, as the barrels will be filled too quickly.

Providence, R. I. Samuel L. Robinson.

# Auxiliary Exhaust Valves on Uniflow Engines

The writer, having read the article by Professor Stumpf in the Mar. 23 issue of POWER, entitled "Recent Development in the Construction of the Uniflow Engine," must take issue with the author on several points, especially the paragraph reading as follows:

**It is wrong in principle to build uniflow engines for condensing service with auxiliary exhaust valves. The short compression is wrong, and just as wrong is the increase of clearance space and surface connected with these valves. Even when used for noncondensing service with steam pressures as used in modern power plants, auxiliary exhaust valves show no gain.**

The reason for the adoption of the auxiliary exhaust valve, as applied to uniflow engines by the Skinner Engine Co., is to allow them to be operated noncondensing without the addition of wasteful clearance spaces which would otherwise be indispensable. It will be of interest to the reader to learn what these valves accomplish.

When the engine is running condensing, these valves are not in operation; but their function on a condensing engine is to relieve the cylinder of dangerous compression if the vacuum should suddenly break. In this event they operate automatically, and the engine continues to run noncondensing and with high economy.

The reader will understand that, with compression extending during 90 per cent. of the stroke and with atmospheric pressure in the cylinder at commencement, as would be the case in a uniflow engine without auxiliary exhaust valves, operating noncondensing, the compression would become so great as to endanger the cylinder. The greatest prejudice against the uniflow engine that European builders have had to overcome was the fact that so many cylinders had been cracked through the central exhaust ports, owing to this excessive compression when the vacuum broke

Two cylinders were thus wrecked at the plant of Vivian & Sons, Hafod Copper Works, Swansea, South Wales; and many other wrecks in Europe have been reported. The writer's understanding is that several cracked cylinders have occurred in this country on condensing uniflow engines not equipped with the auxiliary exhaust valves. Such accidents prove that cylinder relief valves cannot effectually relieve this compression; and in no case that has come under the writer's observation did the compression lift the steam valves a sufficient amount to relieve the cylinder of the excessive pressure.

The clearance volume required for these auxiliary exhaust valves is less than 1 per cent., even on small engines, and about ½ per cent. on large engines. Therefore, the uneconomical effect of these clearances is negligible.

The last statement in the paragraph quoted, "Even when used for noncondensing service with steam pressures as used in modern power plants, auxiliary exhaust valves show no gain," is incorrect.

The professor condemns the use of auxiliary exhaust valves, on account of an increase in the clearance of less than 1 per cent. which is made necessary by their employment, when, if they are eliminated on a noncondensing engine, it is necessary to employ a clearance space amounting to 14.2 per cent. for an engine operating under 120 lb. steam pressure at throttle, with atmospheric exhaust, and 17.3 per cent. if the back pressure at the cylinder is 3 lb. above atmosphere. Still greater clearance must be allowed if superheat is added or if the boiler pressure is lowered.

There are two methods of obtaining this clearance—one by employing separate clearance pockets arranged to be placed in communication with the cylinder when the engine is operating noncondensing, as shown in Fig. 1, and the other by concaving the ends of the trunk piston.

The former method is especially disadvantageous on account of the large additional surfaces introduced. In fact, published economy curves of noncondensing uniflow engines having these clearance pockets show that they are not as economical as many counterflow engines.

By the second method, besides having the handicap of additional clearance, the engine cannot be operated condensing with even fair economy without substituting a flush piston and thereby reducing the clearance.

With either method the clearance volume is fixed for noncondensing operation and, in an existing engine, cannot be varied to suit the changes of steam pressures, steam temperatures or exhaust pressures.

Fig. 2 is a noncondensing indicator diagram from a cylinder of the construction shown in Fig. 1. The amount of clearance required for noncondensing operation with a predetermined steam pressure is shown at the left, in proportion to the stroke of the engine.

That Professor Stumpf realizes this point is proven by the following excerpts from his well known work "The Una-Flow Steam-Engine." Speaking of the uniflow cylinder as applied to locomotive practice, where his own curves show that for a boiler pressure of 242 lb. (which is, of course, much greater than is employed in stationary plants) a clearance of 8 per cent. must be employed for saturated steam and 10 per cent. for superheated steam, and where for 154 lb. pressure he admits that 13.2 per

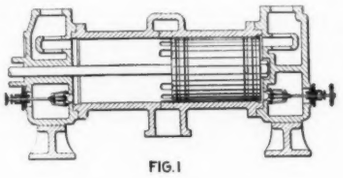


FIG. 1

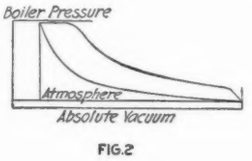



FIG. 2

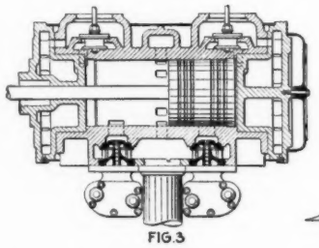


FIG. 3

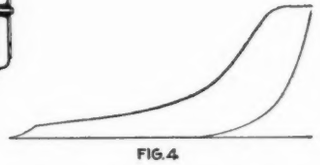


FIG. 4

cent. clearance must be employed for saturated steam and 16.1 per cent. for superheated steam—all on the basis of atmospheric exhaust—he says:

1. **The volume of clearance space should be kept as small as possible.**
2. **The amount of clearance losses depends upon the volume of the clearance.**
3. **The volume of clearance space is dependent upon the pressure and temperature of admission steam.**
4. **In all cases the pressure at the end of compression must not exceed the initial pressure.**
5. **Possible lines of development would be to employ saturated steam, introduced into the cylinder in as dry a state as possible, and superheated only a few degrees, and at a pressure that is at present usual in compound locomotives, so that the clearance space and the loss entailed thereby may be reduced.**

In other words, he practically condemns superheat for noncondensing uniflow locomotives, because it makes necessary still greater clearance. The reader should bear in mind the fact that the pressures carried in modern locomotives are much higher than those employed in stationary plants, and that the clearance, therefore, may be less, but the professor wishes to reduce this clearance still further by employing saturated steam.

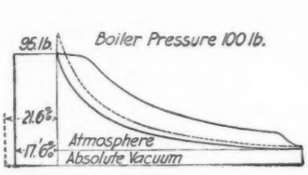



FIG. 5

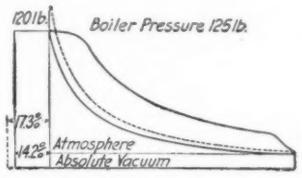



FIG. 6

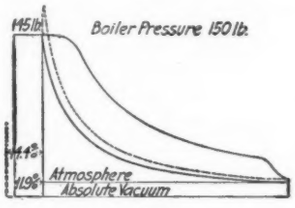



FIG. 7

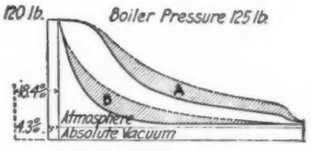



FIG. 8

**6. Other means for improving the action of the uniflow engine for locomotive work are to increase the boiler pressures and, consequently, the compression, so that the clearance volume may be reduced.**

In other words, pressures from 235 to 260 lb. are not high enough for the economical operation of a large-clearance noncondensing uniflow engine.

**7. Comparative tests with uniflow and counterflow engines, working with superheated steam of 11 atmospheres (161 lb. gage), have shown that the advantage rests with the uniflow engine for light and medium loads, but for heavy and overloads the advantage rests with the counterflow engine.**

Here he admits that the large-clearance two-valve uniflow engine is not as economical on heavy loads as the counterflow type. Such is not the case with the small-clearance Universal Unaflow engine employing auxiliary exhaust valves which require the small clearance of less than 1 per cent. which Professor Stumpf objects to, as is proven by the reproduced performance curve of a Universal Unaflow engine operating noncondensing, with saturated steam at 136 lb. initial pressure.

Fig. 3 shows the Universal Unaflow construction, with auxiliary exhaust valves, which have the effect of delaying the compression to that point where it is usual to start the compression in a noncondensing counterflow engine. This construction allows the use of small clearances, even when operating noncondensing, which is not practical with the two-valve uniflow engine. Fig. 4 shows the diagram which it makes.

As a further proof that Professor Stumpf realizes the disadvantages of large clearances, it is only necessary to call attention to the fact that he sought to employ small clearance in a uniflow engine when operating noncondensing, by providing an auxiliary exhaust valve in the piston, which had the effect of delaying the compression beyond the point where the piston covered the central exhaust ports.

The European uniflow engine is, primarily, a condensing engine, for the reason that except in isolated cases all power plants in Europe operate with vacuum. In America, however, the great majority of plants operate noncondensing, and it is due to this fact that the necessity of adapting the uniflow engine to noncondensing work, which meant the employment of auxiliary exhaust valves to delay the compression, appealed to American engine builders before it was seriously considered abroad.

This principle, however, has now been adopted by a prominent German builder, and a description of this engine has been published in the European mechanical press.

One American builder, after having built and thoroughly tested a noncondensing uniflow engine having no auxiliary exhaust valves, now refuses to bid on the uniflow engine for noncondensing service.

The Skinner Engine Co., to determine the relative values of the two types under discussion, has made elaborate tests on both, having built a two-valve uniflow engine for this purpose. The results of these tests were greatly in favor of the engine with the auxiliary exhaust valves, especially when the steam pressure was changed from that for which the clearance in the two-valve uniflow engine was designed.

It was also demonstrated that the capacity of the cylinder was reduced owing to the long duration of compression, namely, 90 per cent. of the stroke; and this reduction of cylinder capacity compelled the employment of a larger cylinder. This in turn would impose greater stresses on the engine and decrease its mechanical efficiency.

The Universal Unaflow engine, which has auxiliary exhaust valves, has obtained mechanical efficiencies in excess of 97.5 per cent., proving the correctness of the principle from a mechanical standpoint.

Professor Stumpf has admitted that, with a noncondensing uniflow engine having no auxiliary exhaust valves, the volume of clearance is dependent upon the steam and exhaust pressures, and that it should be greater for superheated than for saturated steam.

Figs. 5, 6 and 7 show the different clearances required for two-valve large-clearance uniflow engines operating noncondensing, under different steam- and exhaust-pressure conditions. The full vertical line at the left of the diagram indicates the amount of clearance required in proportion to the stroke for different pressures of saturated steam. The dotted vertical line at the left shows the additional clearance required if the engine is to operate against 3 lb. back pressure above atmosphere.

In many plants the steam pressure fluctuates considerably and heating requirements render it advisable to employ a greater back pressure on a noncondensing engine during the cold months; and, inasmuch as it is impossible to vary the clearance in an existing two-valve noncondensing uniflow engine for the changes in steam and exhaust pressures, the engine is not as flexible or as efficient under these variable conditions as one employing auxiliary exhaust valves in connection with small cylinder clearances.

Fig. 8 is a double indicator diagram of a noncondensing Universal Unaflow engine (full line), superimposed on the diagram (dotted line) from a large-clearance uniflow engine having no auxiliary exhaust valves, both engines exhausting against a slight back pressure. The Universal Unaflow engine has 4.3 per cent. clearance, whereas the large-clearance uniflow engine must have 18.4 per cent. clearance for saturated steam.

With the large-clearance uniflow engine, the shaded section *A* must be added to the diagram to offset the loss in area *B* caused by early compression. This loss in area is compensated for only by the addition of more

steam, a later cutoff, a higher release and less expansion.

The writer, however, readily agrees with Professor Stumpf on one point, namely, the necessity for the employment of steam-tight valves. Valve leakage in a noncondensing two-valve uniflow engine would be of more serious consequence than with a noncondensing uniflow engine having auxiliary exhaust ports located between the ends of the cylinder and the central exhaust ports, for the reason that the effect of this valve leakage would be in evidence during 90 per cent. of the compression stroke, and such effect would be to increase greatly the final compression pressure. With the same amount of valve leakage on a uniflow engine having auxiliary exhaust valves, the effect of leakage operates to increase compression during only 25 or 30 per cent. of the stroke.

Nevertheless, it is well to eliminate even this valve leakage, and this has been accomplished in the case of the Universal Unaflow engine by the adoption of a self-expanding poppet valve.

A. D. SKINNER.

Erie, Penn.

## Convex and Concave Drum Heads

The discussions during the past year in POWER, with regard to the allowable pressure on convex drum heads, were of a purely technical nature, in which the aim was to discover the proper factor of safety, having in view explosions due to failure of convex heads near the root of the flange, by the tearing out of a circular section. It was pointed out by one writer that the factor of safety, 5, assumed in all rules in this country was entirely too low and that a higher factor, at least 8.33, should be used. F. G. Gasche, in discussing the subject advocated a factor of safety of 15.2, based on a thorough analysis of the stresses set up on such heads. Since that time another disastrous failure has occurred, resulting in heavy property damages and personal injuries, that also adds to the proofs of the fallacy of the factor of safety of 5.

In this connection I would mention recent tests to destruction of two concave heads that failed under hydrostatic tests. The vessels were tanks or drums, having the following dimensions: Diameter, 114 in.; shell plate, 1 in.; long seams, butt, double-strapped, triple-riveted, with an efficiency of 80 per cent.; tensile strength of heads and shell, 55,000 lb.; thickness of heads, 1⅛ in.; heads bumped to 114-in. radius and single-riveted to the shell. The required pressure was 150 lb., which gave a factor of safety of 5 in the shell. The sketch shows how the heads were put in, one being concave and the other convex. The heads were from the same block, being duplicates in each case. Moreover, the drums were new and were tested where built.

The first drum failed at about 210 lb., the failure occurring in the *B* head near the flange and also by tearing the shell near the head seam. The second drum failed by two ruptures in the head, the shell developing no defects, but the failure took place at 150 lb.

Commenting on this, I am of the opinion that the head in the first tank was harder or stiffer than in the second, inasmuch as the shell was injured. In other words, in the second case the head sprung in the center enough to cause it to rupture at a point relatively near the flange, and in the first one the head was strong enough to force a rupture in the shell adjacent to the head.

It seems to the writer that tests to destruction must be accepted as conclusive practical results on which to base deductions to determine the safe working pressure. To this end let us compare the results with the present rule as follows:

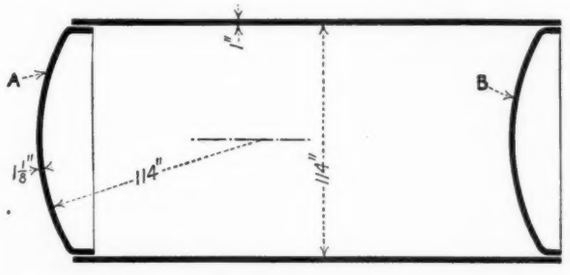
$P$ = Working pressure allowed;
$t$ = Thickness;
$T.S.$ = Tensile strength;
$R$ = Radius, or ½ diameter of drum;
5 = Factor of safety.

Then

$$P = \frac{t \times T.S.}{R \times 5} = \frac{1\frac{1}{8} \times 55{,}000}{57 \times 5} = 217 \text{ lb.}$$

as respects the *A* head, and 0.6 of this, or 130 lb. on *B*. Using the factor 8.33, the pressure allowed on *A* would be 130 lb., and on *B*, 79.8 lb. This gives a real factor of about 2, based on the 150 lb. at which failure occurred.

By Mr. Gasche's factor, *A* would be allowed 71.4 lb. and *B* 42.8 lb., making the real factor about 3 on the concave head. In all the above calculations the tensile strength is taken at 55,000 lb. per sq.in.




SECTION THROUGH DRUM THAT WAS TESTED TO FAILURE

Granted a test to destruction is accepted as a practical method in determining the safe allowable pressure, and assuming that there should be a factor of 5 to safeguard against accident during the presumed life of a vessel, then the pressure allowed would be one-fifth the bursting pressure, or 30 lb.; as regards the weakest part, namely, the concave head.

In discussing these failures of concave heads we have the empirical rule of allowing six-tenths the pressure allowed on convex heads, but one can readily see that the stresses set up are vastly different from those on convex heads, especially when it is considered that a vessel subjected to internal pressure tends to assume a spherical shape, and an inquiry as to the foundation of this rule is in order.

Along this line it seems to the writer that our technical colleges might well take up the matter of convex and concave heads and make a thorough inquiry and give the results to the public. They have facilities for doing research work of this sort that are beyond the practical man in the field. Surely the situation warrants investigation, where the factors of safety vary from 5 to 15, and only in the latter is the efficiency of the (usually) single-riveted joint taken care of, and such joints are only about 50 per cent. as strong as the plate.

D. HOGAN.

New York City.

## Practical Pump Slippage Test

In a large pumping station the amount of slippage was based upon the difference between the calculated displacement and the meter reading. At one time the station slippage recorded was abnormally high. As the combined output of all the pumps entered the same main, some test was necessary to locate the responsible pump.

To do this the station was operated for a specified time with each pump shut down in turn. The pump that was out of service when the best slippage record was made, was identified for special test. The suction side manhole plates were removed, and water was then bypassed to the discharge chamber and allowed to accumulate a pressure there. Large quantities of water rushed from the suction manholes. This indicated that no mistake had been made and resulted in a decision to revalve the pump.

EDWARD T. BINNS.

Philadelphia, Penn.

## Explosion of Hot-Water Tank

Power plants are not the only scenes of disastrous explosions. On Oct. 20, 1914, at about 10:30 p.m. on the premises at 18 McCann St., Ilion, N. Y. a hot-water tank exploded, with the results shown in the reproduction of photograph taken the morning after (Fig. 1).

An investigation showed that a system of installing a pressure-reducing valve between the street main and the dwelling is in vogue in the village, as shown in the sketch,

FIG. 1. CONDITION OF PREMISES AFTER EXPLOSION

Fig. 2. The reason for this is that the pressure in the mains is upward of 150 lb. per square inch at many points. Houses on the slopes or tops of hills do not have extraordinary pressure, but the majority of dwellings and business places have this system with a reducing valve, as shown in Fig. 3.

As such valves are absolutely nonreturn, when one is placed in a dwelling without a safety valve of some kind on the house side, there is no automatic means of relieving the pressure if it goes beyond that for which the reducing valve is adjusted. The only means of release is opening some of the various taps by hand.

The family in this case left home to spend the evening with friends, leaving the gas jet burning under the heating coil attached to the hot-water storage tank. This allowed the pressure at the weakest spot to reach the bursting point. Without the reducing valve in the system the pressure would have relieved itself into the city main. That the stop-cock next to the street was open was proved by the fact that the first man to arrive on the scene after the explosion closed it to prevent further flooding of the premises.

In this case the weakest point was the lapwelded type hot-water tank in common use, which let go along the longtitudinal seam from end to end. The force was great enough to tear the seam apart, turn the sheet out flat

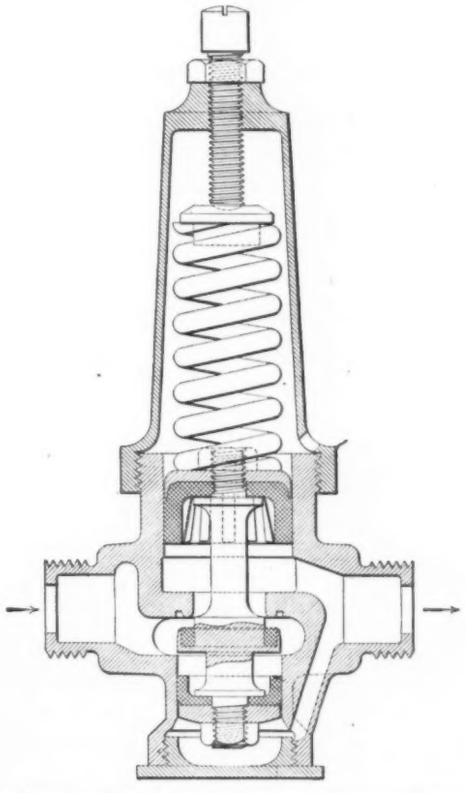


FIG. 3. TYPE OF REDUCING VALVE USED

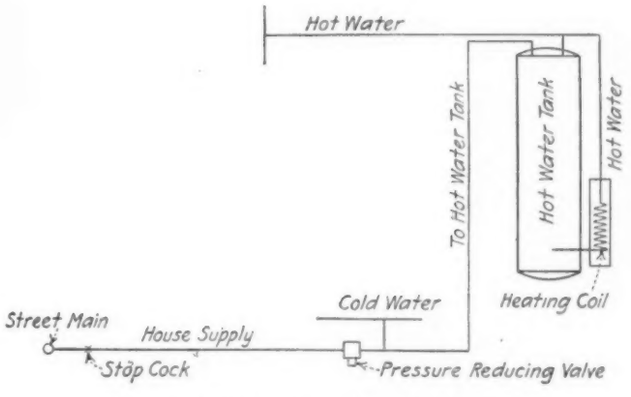



FIG. 2. DIAGRAM OF PIPING SYSTEM

and double it over from end to end as usually happens in such cases.

This accident shows that the authorities should insist on the use of safety valves where reducing valves are installed or should not allow the use of the latter at all.

The wrecked dwelling has been rebuilt and the water system installed in this same place without the reducing valve. There are many other places where the old system is in use, and no attempt has been made to correct the fault in such installations. Other municipalities have the same conditions, and other explosions have occurred from the same cause. All should take warning, for the continuation of this system will mean more loss of property and possibly loss of life, which will amount to criminal negligence on the part of those who have the power to correct this condition and fail to do so.

HUBERT E. COLLINS.

Utica, N. Y.

## Testing for Open Circuit

In the Feb. 9 issue, page 195, there was described a practical method of testing for an open circuit in a starting resistance. This is to close the line switch, throw the starting-box arm on the first contact, and then bridge between the contact buttons with a piece of metal, such as a screwdriver. Of course, the motor will start if the metal bridges across the open-circuited part of the resistance.

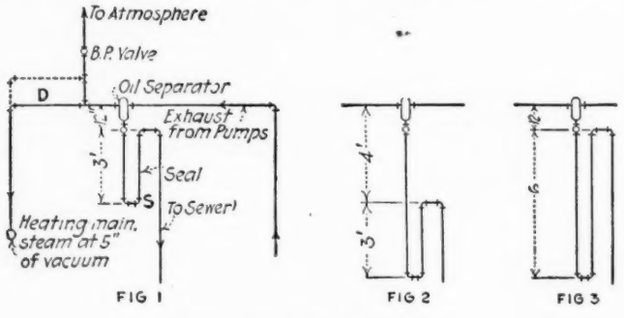
The scheme is possible and no doubt has been used safely by the writer of the article. Nevertheless, it seems dangerous. An open circuit means full line voltage at the break, under the conditions specified. Would one use a screwdriver to close a 550-volt circuit? When the test is made, practically the resistance is in circuit, but should the screwdriver slip and bridge across the contacts at opposite ends of the box, injury to the operator's eyes might result.

R. E. PLIMPTON.

Brooklyn, N. Y.

## Oil Separator Failed to Work

An article in the Nov. 3, 1914, issue of POWER, page 650, by T. W. Reynolds, under the above heading, is faulty in the application of the figures arrived at as a solution for the failure of the separator drain to work.




SEAL FOR OIL SEPARATOR

Mr. Reynolds has treated his problem as though the vertical drain pipe from the bottom of the separator dipped directly under the surface of water and oil in a cistern open to the atmosphere. His solutions are correct for this condition, and the dimensions given will work out in practice.

*Interesting in this connection are the letters on "Trouble with Oil Separator," Feb. 9, page 207, and Mar. 9, page 344.

For the arrangement shown in the diagrams, however, the dimensions given are incorrect, for the following reason: With the siphon type of seal (this is the type shown in the sketches) the liquid discharged from the separator must all pass over the crest of the siphon, and the limiting height for proper discharge will be the distance between the crest of the siphon and the allowed height of the liquid in the separator, not, as Mr. Reynolds has erroneously assumed, the sum of the distances from the crest of the siphon to the level of the liquid in the separator and the length of the riser leg of the siphon. This being the case, it is evident that the arrangement shown in Fig. 3 is no better than that shown in Fig. 1 for removing drips from the separator under a vacuum of 5 in.

To fulfill the conditions given in Mr. Reynolds' article the distance between the crest of the siphon and the bottom of the separator would need to be about six feet instead of four feet as shown in Fig. 2. This gives a column of liquid above the crest of the siphon equal to 6 times 0.434 lb. per sq.in., or 2.604 lb. per sq.in., which is slightly in excess of 5 in. of vacuum, which equals 2.46 lb. per sq.in.

With the six-foot dimension instead of the four-foot dimension shown in Fig. 2 of Mr. Reynolds' article, the siphon seal will properly remove all the condensation and drips collected by the separator, and if the riser leg of the siphon were made six feet also (keeping the crest of the siphon six feet from the bottom of the separator, as before outlined) the seal would properly take care of the drips collected by the separator under all conditions of pressure in the heating system from 5 in. vacuum to about 2½ lb. back pressure, that is, pressure above the atmosphere.

R. N. ROBERTSON.

Denver, Colo.

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The only thing that is wrong with Mr. Reynolds' proposed remedies is that they are no better than the original, and besides, the figures he uses are for water at 39 deg. F., and not for cylinder oil, although water of condensation will be caught by the separator also.

The original layout has two faults. One is that the top of the seal is too near the same level as the bottom of the separator. Another is that the seal is not deep enough. As the pressure in the main will sometimes be atmospheric, the depth of the seal should be such as to make it safe at atmospheric pressure.

Cylinder oil weighs about 0.39 lb. per square inch per foot of column. I would make the height of the seal rather more than

$$\frac{2.46}{0.39} \times \frac{1}{2} = 3.15\ ft.$$

After being in operation for a few minutes the discharge side of the seal will be full of water and this column of water 3.15 ft. in height will counterbalance about 3.5 ft. of oil. I would therefore place the top of the seal

$$\frac{2.46}{0.39} + (3.5 - 3.15) = 6.65\ ft.$$

below the bottom of the separator.

His sketch No. 3 would not do for any degree of vacuum.

R. MCLAREN.

Medicine Hat, Can.

# Will Quizz, Jr.

*SYNOPSIS—Will Quizz, Jr., has a lot of questions to ask Chief Teller about the hydrometer and the various graduations on it.*

"Have you tested the brine in the cooling tank lately, Will?"

"Yes, Chief, the specific-gravity gage shows 1.206. I took along the other gage, the Baumé, and that showed 26. What do these figures mean?"

"To begin with, get the object of the test clearly in mind. What do you test the brine for, anyway?"

"To find out whether it has enough salt or calcium chloride in it or not."

"Yes, but couldn't you tell that in some other way? Tell me how."

"I could take a graduated flask and draw off a certain amount of the liquid, evaporate the water out of it and weigh the salt it contained, but that would be a lot of bother."

"Suppose, then, you put the same amount of salt into the same quantity of water and weigh the mixture carefully, you would find that for a certain degree of saltness the weight would be the same every time and for a greater or less degree the weight would vary accordingly. This method would not be convenient, either, because care must be taken to get just the correct quantity each time and weigh it carefully. By the way, did you ever think of why they always use those peculiar-shaped bottles or flasks, with long slender necks on which there are graduations, in the laboratories? The idea is to fill to a certain mark in a slender part for accuracy, because a few drops more or less will change the level in a slender tube a lot, while if the mark were on the large body of the flask a difference of considerable magnitude would hardly be noticeable. This same feature applies to the hydrometer, which will be referred to later.

"Now, suppose you used the same flask, but instead of putting the fluid into it, you put certain weights inside of it and put the flask into the brine. It would sink to a certain depth, displacing an amount (measure) of liquid exactly equal to the volume or measure of that part of the flask which was submerged and in weight equal to the total weight of the flask and contents. The level to which it settled could then be marked on its neck so that it might be used again as a test gage for other brine of the same density into which it would settle to the same mark. (This is where the long slender neck of the hydrometer enters in, as referred to before. As the part is slender, it would have to be submerged to a greater extent to displace a small amount of liquid, and the graduations would be farther apart and more easily read.) This method would be more convenient than measuring out a given quantity and weighing it, or evaporating it in the way you just spoke of. Then by a series of tests you might construct a scale on the neck of the flask, so that you would know the density or saltness of the fluid by the depth to which the flask settled and would know by that scale how much salt there was to a cubic foot just as well as if you had gone to the trouble of evaporating the water out. If, after getting your flask nicely graduated for brine, you should put it into pure distilled water it would be likely to sink 'head over heels' unless the neck was very long. This is because the fresh water is less dense and the same vessel would sink deeper or entirely."

"Yes, Chief, I have always understood that objects which will sink in fresh-water streams or lakes will sometimes float in the ocean. So this is the same thing, is it?"

"That's the idea, Will. Then if you were making a hydrometer you would want a fresh-water mark on it as a means of comparison. That's just what Baumé, Twaddell, Beck and others did in constructing an arbitrary scale. The way Baumé first arrived at his scale was, the instrument was submerged in water, by means of the mercury placed in the bottom of the glass, to a certain point which was marked zero. If the instrument was to be used to determine the density or specific gravity of fluids heavier than water, it would be loaded so that it would sink in distilled water almost to the top of the tube, because with the same amount of weight it would not sink so far into the heavier fluid, therefore the graduation was downward from the zero mark made at the surface of the water. The instrument was then put into a solution of 15 parts of salt and 85 parts of water. The point to which it would sink in this solution was marked 15, the distance between these two points was then divided into 15 equal parts, and the graduation was continued beyond 15 in equal divisions. This constitutes the Baumé scale for liquids of greater specific gravity than water. On the other hand, if the instrument was to be used for fluids lighter than water a different scale was used. Baumé used for the zero point the position of the instrument in a solution of 10 parts of salt and 90 parts of water, and for 10 its position in distilled water, and divided this distance into 10 deg., and continued the graduation to the top of the scale.

"There is a tendency now to discard all arbitrary scales in favor of those which read in terms of specific gravity without the necessity of interpolation. The Baumé scale in its time was a very important development, but since the specific gravity of the common fluids has now been established so that the instruments marked with a scale showing the specific gravity by direct reading are preferable. It will be interesting for you to look up a table showing by comparison the Baumé scale and the specific-gravity scale."

"Yes, Chief, such a scale does not mean much, to me at least."

"The specific-gravity scale shows by direct reading the weight of the fluid as compared with the same volume of pure distilled water at 65 deg. That is, the brine in this case is 1.206 times as heavy as water (62.35 per cubic foot), the brine would be $62.35 \times 1.206 = 75.194$ lb. per cubic foot. Any reading of specific gravity multiplied by the weight of water per cubic foot will give the weight per cubic foot of the fluid in question. The reading on the Baumé scale in so called degrees must be transposed by reference to a table to become intelligible."

"The term specific gravity gets me twisted somehow, Chief. See if I have it right. Gravity, or weight, refers to the action of the law of gravitation acting on a given

substance. Then the word specific added makes the weight refer to and compare specifically with some other recognized substance, so that the specific gravity of a substance is its weight compared with distilled water, bulk for bulk. Is that right, Chief?"

"Yes The weight of an object as shown by the scales is independent of its bulk or volume. Its density is its weight per unit of volume (1 cu.ft. for example). Its relative density, or specific gravity, is its weight per unit of volume (as before) specifically compared to the weight of the same volume of a predetermined standard (pure distilled water at maximum density for heavy things is generally used and hydrogen for gases)."

"Another thing puzzles me, Chief. If a body which sinks is put into water it will displace a certain amount of the water which, if allowed to overflow and is measured, will occupy the same space as the body which has been put into the water, but they do not weigh the same. In the other case if the body placed in the water floats, the water which is displaced by it will be of the same weight, while the volume of water displaced will be equal to only that part of the body which was immersed, yet the specific gravity of each is given in the table. How is that determined?"

"The specific gravity of the body heavier or more dense than water is obtained by sustaining some of its weight, or that part of its weight greater than the weight of the water displaced by it, by means of scales with the proper weights on the opposite side; then in every case the weights so required plus the weight of the water displaced will equal the weight of the body being tested, while the ratio of the weight of the body in air and in the water is its specific gravity.

"Density is defined as 'the ratio of mass to volume.' It was considered worth while by the International Congress of Physicists at Paris in 1900 to pass a resolution defining this quality as stated above, because the term is so frequently misused by writers on scientific subjects. Density and specific gravity are by no means the same, although one is proportional to the other. The specific gravity of any substance is defined as the ratio of its density to the density of water.

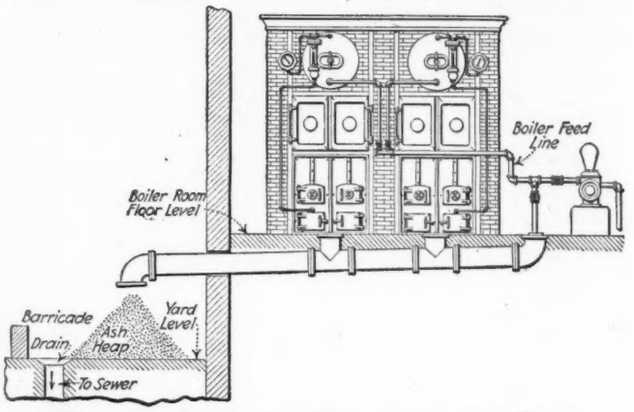
"This old problem may interest you, as it is along the same lines: A canal aqueduct is capable of sustaining ten tons per running foot, the structure filled with water weighs seven tons per foot, and a boat weighing four tons per foot is to pass through the canal. The question is, will the structure sustain the weight when the boat is in the canal?"

## Ash Handling by Flushing

An interesting way of handling ashes was observed during a visit among some small plants where the number of boilers did not warrant the use of conveying machinery.

In one plant, a part of which is shown in the sketch, the yard level was below the boiler-room floor. To convey the ashes to this level 6-in. terra cotta soil pipe was laid as shown. The elbow at the extreme right was blanked off and a 2-in. pipe inserted. Water for flushing the ashes out of the pipe is admitted through this line. In this plant water from the feed-pump discharge was used.

The large pipe should be inclined considerably and large clinkers should be broken before entering the tees.




How the Ashes Are Flushed to the Yard

## Centrifugal Crude-Oil Pump

By A. B. Morrison, Jr.

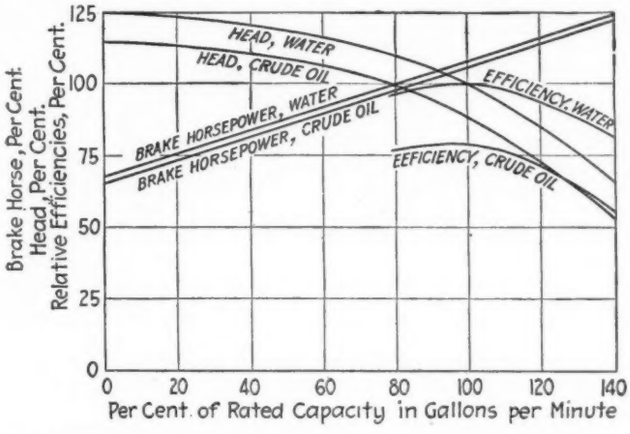
In handling liquids other than water with a centrifugal pump it is necessary to study closely the characteristics of the liquid to be pumped and its behavior under similar operating conditions in order to estimate even approximately what can be expected in the way of power and speeds to produce certain results. A centrifugal pump was used for handling crude oil, replacing a direct-acting steam pump which had been used for the same service. The results obtained were so much at variance with what had been expected that some further tests were made to ascertain wherein the difference lay, and as a result some interesting data were obtained.

The centrifugal pump was designed to deliver 1600 gal. per min. against a head of approximately 60 lb., the head being due almost entirely to the friction in a long line of pipe, the static elevation being slight. The pump was of the single-stage, horizontal split-casing type, direct connected to a steam turbine running at approximately 2000 r.p.m. The guarantees as to head, capacity and steam consumption were made on the basis of pumping clear water. On the basis of such data as were available to the customer, it was assumed that the friction of the crude oil through the discharge pipe would be approximately the same as that of an equal amount of water. It was, therefore, considered that the pump should show the same efficiency, approximately, pumping oil as water. The crude oil had a specific gravity compared to water of 0.865 to 1.

A test was run on the outfit as installed, by measuring the steam to the turbine with a steam-flow meter, the oil by measuring the total amount taken from the tank and dividing by the total minutes run to get the average quantity pumped per minute, and the pressure by means of gages on the suction and discharge, correcting for the difference in level between these gages and the center line of the pump. The speed was recorded at frequent intervals, so that a good record of the pump performance was obtained. Allowing for possible errors in the readings and giving the outfit the benefit of every doubt, the steam consumption, as shown by two different tests, was so much greater than that guaranteed that it was evident something was wrong. A test of the turbine and pump separately, the latter pumping water, showed that both met

their respective guarantees as to capacity and efficiency, so there was no apparent reason why the combined outfit should not give the results anticipated if the assumption as to the power required for handling crude oil were correct.

To determine how much difference existed between pumping crude oil and water a series of tests was run on a smaller motor-driven pump, as it was not possible to run the turbine outfit on anything but oil. The pump available for the tests was a small single-stage side-suction one direct-connected to a three-phase motor. It was old and not in good condition. The suction was under a slight pressure and the discharge was into a tank, the pressure on the discharge being varied by manipulating a valve. By means of a float on the tank the average quantity pumped in gallons per minute was determined for five-minute intervals. Gages on the suction and discharge gave the pressures. The gages on the suction were, apparently, not accurate, so that the readings were not wholly reliable. At the time the test was made no wattmeter was available, so it was necessary to read the current and voltage in one phase only and assume an arbitrary constant for efficiency and power factor in calculating the horse-




PUMP PERFORMANCE WITH WATER AND WITH CRUDE OIL

power. For this reason especially, the curves are not correct, and it should be understood that the relative brake horsepower, head and efficiency are approximate. The general forms of the curves are correct. Since the essential idea was to determine the relative behavior of the water and the crude oil, it was considered that the tests answered the purpose.

An inspection of the curves shows the marked increase in the efficiency of the pump when handling water. The brake horsepower was approximately the same in both cases, though the weight of the oil pumped was considerably less and the head generated by the pump less. Through the pump the velocity was very high as compared with that in an ordinary line of pipe, and this high velocity of the oil caused, on account of the viscosity of the latter, the loss in efficiency. This is shown by the head curve. Theoretically, the head in feet should be the same regardless of the liquid pumped, but part of the head generated is lost in the pump because of the greater work required to get the oil through the impeller and casing.

One fact not shown in the curves, but brought out in the operation of the large pump, is that the assumption that the pipe friction is about the same for crude oil and water at the usual pipe velocities is quite correct. The crude oil was pumped through the pipe line in the calculated and desired quantity, and while the required pressure, measured in feet, was slightly greater than that for water, it did not differ from that estimated enough to occasion any trouble. When the velocity is as high, however, as is necessary to get the liquid through a centrifugal pump, the greater viscosity of the oil causes the very marked increase in the friction and power required.

In view of the results obtained with the outfit described, it is a question whether it would not have been better to install a compound direct-acting steam pump. The conditions were not favorable for a steam turbine, as the steam pressure was only 70 lb. gage and the exhaust was to the atmosphere. The pump also ran somewhat slower than the most economical point of the turbine. There is, of course, the argument for greater simplicity in the turbine-driven outfit, both in prime mover and pump, but the ordinary direct-acting steam pump is not likely to give trouble and in the present case the steam consumption would have been no greater and the first cost probably less.

## Steam-Turbine Drive for Steel Mills*

The Carpenter Steel Co., of Reading, Penn., furnishes an interesting example of how increased power can be obtained, while reducing the fuel bill and the number of boilers in service. This plant is, moreover, notable as being the first in America, and the second in the world, to apply the steam turbine to the driving of rolling mills through the medium of mechanical speed-reducing gears.

The company does a general merchant trade, consisting of high-grade tool steels, projectiles and special steels for such uses as safety-razor blades, springs, and the like. The two-stand, 18-in., three-high roughing mill now driven by turbine was formerly driven by a 36x36-in. simple slide-valve engine, operated condensing, while the 10-in. and 8-in. finishing mills were driven by belt from a cross-compound, 22& 40x48-in. engine, exhausting into a jet condenser, which gave a vacuum varying from 18 to 24 in.

Besides the main power units for the mill, there were in the immediate vicinity several service and boiler-feed pumps and air compressors, the exhaust of which was partially utilized in an open feed-water heater, the surplus escaping to the atmosphere. While no data were obtained regarding the steam consumption when using this equipment, it was necessary that five boilers of a nominal total rating of 1000 hp. be operated continuously.

With a view to reducing power costs, several alternatives were considered. The simple engine driving the roughing mill could have been replaced by a modern compound engine and a modern high-vacuum central condensing plant for the two engines put in. But because of the fluctuations of load on the engines and the small average horsepower required, also because of the first cost of the equipment and the moderate economy, this scheme was not adopted.

The second alternative was the installation of a low-pressure turbo-generator operating on the exhaust of the compound engine to supply current to a motor driving the roughing mill. This did not appear attractive, as it involved a large investment in turbine, generator, switchboard, transmission lines, motor, starters etc., and nearly all the current produced would have been consumed by the roughing mill.

The third plan considered, and the one ultimately adopted, was the installation of a low-pressure turbine to drive the roughing mill. This proposal involved the use of speed-reducing gears, a new expedient for this work, but one which had already been satisfactorily used by James Dunlap & Co., Calderbank Steel Works, near Glasgow, Scotland, where a mixed-pressure turbine, developing 750 hp., drives a three-high, 28-in. plate mill. The speed reduction of the Calderbank mill is made in two steps—first from 2000 to 375 r.p.m. and then to 70, by means of double-helical gears of the rigid-frame type.

As compared to the 15 to 20 per cent. loss of energy in

*From a booklet issued by the De Laval Steam Turbine Co.

the electrical method of driving, the gear loss is not more than 1½ to 2 per cent., besides which it costs less, occupies much less space, is simpler and requires less attention.

For the operation of the roughing rolls at the Carpenter steel plant, an average of about 350 hp. was required, at a speed varying from 60 to 100 r.p.m., while for the greatest efficiency and simplicity of construction a turbine of this capacity should run at about 5000 r.p.m. To secure the reduction of 50 to 1, a two-step reduction gear was adopted, the gear and turbine being mounted on one base plate and the complete unit so located that the shaft of the slow-speed gear is in line with the shaft of the engine which the turbine replaces. The 26-ft. engine flywheel, weighing 47,600 lb. and the engine shaft and bearings were left in place. In order to avoid interruption of work while making the change and also to provide against any possible interruption of service thereafter, the engine was left intact and only the connecting-rod removed, a distance piece, which serves also as one of the flanges of the flexible coupling on the low-speed gear shaft, being bolted onto the crank disk. After the turbine was installed it was thus possible to operate it without load in order to try out the installation, during which period the engine continued to drive the mill and practically no time was lost in changing from engine to turbine drive.

The turbine, which is of the mixed-flow type, was built by the De Laval Steam Turbine Co., and contains eight pressure stages, the first pressure stage consisting of two velocity stages. The low-pressure steam, that is, the exhaust steam from the engine, is admitted at the third pressure stage and expands through the remaining stages to exhaust pressure.

The turbine is designed to operate under four different steam conditions: When receiving engine exhaust at a pressure of 3-lb. gage, and when exhausting into a vacuum of 27 in., it is to develop 350 hp. at speeds corresponding to 70 to 100 r.p.m. of the mill shaft, under which condition it is guaranteed to take not more than 26 lb. of steam per brake horsepower per hour, as measured at the end of the second gear reduction. It is also to be able to carry the normal load of 350 hp. when using steam at 120 lb. pressure gage and exhausting to a 27-in. vacuum, under which conditions it is guaranteed to take not more than 17½ lb. of steam per brake horsepower per hour. When receiving both high- and low-pressure steam and exhausting to vacuum, the turbine is to be able to carry 600 b. hp. continuously. It is also to be able to carry a load of 600 hp. on high-pressure steam only, as when the compound engine is not running. Under these conditions it is guaranteed to take not more than 15.7 lb. of steam per brake horsepower per hour.

The turbine is also to be able to carry normal load noncondensing when receiving steam at 120 lb. and exhausting to atmosphere, making it possible to inspect or repair the condenser or circulating pump without interfering with the operation of the mill.

The roughing mill is three-high and consists of two stands of 18-in. rolls. It is manually operated, two men being employed on each stand. The reduction of a 4x4-in. billet, 17.6 in. long, weighing 80 lb., to an oval 1¾ in. wide is performed in thirteen passes, occupying 41⅔ seconds.

Owing to the fact that the finishing mill cannot take high-carbon steel as fast as the roughing mill can supply it, only one billet is usually in the latter at one time, but when rolling low-carbon steel, two billets are in at once. When handling one billet no speed variation is perceptible on the tachometer, which is permanently attached to the 600-r.p.m. shaft, while when handling two billets the speed drops about 2 per cent. The fact that there is no drop of speed with one billet in the mill shows that the heavy flywheel is not required for power storage under such conditions. By setting the governor for a greater drop in speed before the high-pressure valve opens, the power-storage capacity of the flywheel can be utilized and unnecessary use of high-pressure steam avoided.

The compound engine, operating noncondensing, ordinarily supplies steam to the turbine under 3 to 3½ lb. back pressure, the pressure being regulated by a multiport safety exhaust outlet valve. The roughing mill, however, is not required at certain times, the steel being taken directly to the finishing mill, at which times the valves between the engine exhaust and the turbine, and between the turbine exhaust and the condenser can be closed and the valve between the engine exhaust and the condenser opened, permitting the engine to operate condensing.

In the low-pressure supply line to the turbine is placed a receiver steam separator. To insure dry steam, a coil of 1-in. copper pipe, inserted in the 12 ft. of 10-in. pipe between the separator and the turbine, is kept filled with live steam and drained by a steam trap. Adjacent to the low-pressure throttle valve of the turbine is a multiport flow valve, designed to prevent the vacuum from backing up from the condenser through the turbine into the engine exhaust line. Without this valve there is always a possibility of air being drawn in through leaks in the exhaust line and through piston-rod and valve-stem packings of the engine at times when there is little or no exhaust steam available. Such air would interfere with the operation of the condenser.

The condenser, which is of the multijet type, is located just outside of the engine room and is protected by a 10-in. multiport atmospheric relief valve. With a water temperature of 72 deg. F., the condenser maintains a vacuum of 28.2 in. with the barometer at 29.78 in. As there was a surplus of exhaust steam for operating the circulating pumps, and also because of the desirability of a simple, reliable condenser, this type was considered best.

The circulating water for the condenser is supplied by a centrifugal pump driven by a mixed-flow geared turbine of the velocity-stage type with two sets of nozzles and designed to operate either with steam at 120-lb. gage or with steam at 3-lb. gage, exhausting to a 27-in. vacuum. It exhausts through a 10-in. pipe to the same condenser that serves the main unit. The speed-reducing gears permit both turbine and pump to run at the best speeds for economy, viz., 5000 r.p.m. for the former and 1500 r.p.m. for the latter. The turbine can be started noncondensing and will carry full load with high-pressure steam alone, thus providing both for starting the pump before the condenser is in operation and for carrying its full load without taking low-pressure steam. Surplus exhaust steam not used by either turbine passes to the open-feed water heater, which also receive the exhaust from pumps, air compressors, etc.

The cost of the installation, consisting of the turbine, reduction gear, condenser, piping, circulating pump, valves, etc., erected complete, was not far from $25,000, and owing to the better economy secured through its use, it has been possible to reduce the boilers in operation from 1000 hp. to about 600 hp. This indicates a saving approaching $15,000 per year.

## Cause of the "San Diego" Explosion

The cause of the low water which allowed the boiler aboard the U. S. armored cruiser "San Diego" to become overheated and explode on Jan. 21, it is learned, was the top of a bucket strainer in the feed tank came off, dropping to the bottom of the strainer, partly closing the opening of the suction pipe to the feed pumps. One-half of the supply of feed water was thus shut off and as water was being carried rather low and the engines running at full power, it was impossible to get sufficient water to all the boilers, with the result that five boilers were badly overheated, when the explosion occurred.

The "San Diego" (originally the "California," built in 1907) is the flagship of the Pacific fleet that had just completed a four-hour speed trial. Her triple-expansion engines indicate 30,000 hp., and are capable of developing a speed of 22.5 knots. The boilers are of the B. & W. marine type (forced draft), with 1592 sq.ft. of grate surface, and 70,000 sq.ft. of heating surface. She carried a crew of 822 men, nine of whom were killed as a result of the explosion.—A. P. Connor, Washington, D. C.

## Foreign Trade Opportunities

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|---|---|
| Brass and phosphor bronze wire | No. 15,678 |
| Centrifugal pumps | No. 15,714 |
| Concrete mixers | No. 15,744 |
| Electric motor | No. 15,699 |
| General agency | No. 15,691 |
| General agency | No. 15,740 |
| Iron and steel | No. 15,757 |
| Machinery and tools | No. 15,743 |
| Machinery | No. 15,727 |
| River dredge and pumping plant | No. 15,675 |
| Steel forgings, castings, etc. | No. 15,673 |
| Sugar machinery | No. 15,701 |
| Tools and technical appliances | No. 15,692 |
| Vacuum cleaners | No. 15,718 |
| Vacuum cleaners | No. 15,750 |
| Wire, iron and steel bars, paint colors, etc. | No. 15,677 |
| Wire machinery | No. 15,690 |
| Wrought-iron fittings | No. 15,676 |

Addresses and detailed information may be obtained from the Bureau of Manufactures, Washington, D. C., and its branch offices, as follows: New York, Room 409, U. S. Customhouse; Boston, 752 Oliver Bldg.; Chicago, 629 Federal Bldg.; St. Louis 402 Third National Bank Bldg.; Atlanta, 521 Post Office Bldg.; New Orleans, 1020 Hibernia Bank Bldg.; San Francisco, 310 U. S. Customhouse; Seattle, 922 Alaska Bldg.

## Recent Court Decisions

Digested by A. L. H. STREET

**Fires Set By Traction Engines**—One operating a steam roller or traction engine along a street or highway must use that degree of care to avoid setting fire to adjoining property which an ordinarily careful person would use under the same circumstances. And if a law or ordinance requires such engines to be equipped with spark arresters, the owner is liable for loss directly attributable to failure to comply with the requirement. These rules were lately announced by the Delaware Superior Court in the case of Cecil vs. Mundy, 92 "Atlantic Reporter" 850.

**Classification of Fuel-Gas Rates**—A public-service corporation engaged in furnishing gas for heating, lighting and power purposes may lawfully classify its rates according to the nature of the service afforded, as well as the quantity of gas furnished, if the classification is not unfair or discriminatory against other classes of consumers. The West Virginia Supreme Court of Appeals has just handed down this decision in the case of Elk Hotel Co. vs. United Fuel Gas Co., 80 "Southeastern Reporter," 922. But the court holds that the same rate and service must be offered alike to all consumers similarly situated and provided with the same character of equipment.

**Impairment of Franchise Rights**—After a power company has acquired a right to use streets and highways for the maintenance of transmission lines under existing constitutional and statutory provisions, the right cannot be impaired by a subsequent constitutional amendment. Hence, the provision incorporated into the Michigan constitution in 1909, to the effect that public-service corporations shall not be permitted to use the streets of a city without obtaining a franchise from the city, cannot be deemed to impair a previously acquired right to maintain a line along certain streets. (Michigan Supreme Court, City of Lansing vs. Michigan Power Co., 150 "Northwestern Reporter," 250).

**Washroom Law Sustained**—In 1913 the Illinois Legislature enacted a law which, in effect, requires that owners and operators of coal mines, steel mills, foundries, machine shops, etc., provide adequate washroom facilities to enable employees whose work in such employments causes their persons or clothing to be covered with grime, dirt or perspiration to such an extent as to render their remaining in that condition unhealthful or offensive to persons with whom they come in contact in leaving their work, to change their clothing and wash before leaving. In the case of People vs. Solomon, which was recently before the Illinois Supreme Court, the validity of this regulation is upheld. (106 "Northeastern Reporter," 458.) Doubtless it will be construed as extending to the employment of stationary engineers and firemen in the employments covered by the law. In fact, the decision seems to hold that it covers all employments where "employees become covered with grease, smoke, dust, grime and perspiration" to the extent that their remaining in that condition would be unhealthful or offensive to the public.

**The Engineer as an Expert Witness**—The qualifications of an engineer to testify as to the availability of means to prevent injury to adjoining property in the cleaning of locomotive boilers were under consideration recently before the Pennsylvania Supreme Court in the case of Vile vs. Pennsylvania R.R. Co. (91 "Atlantic Reporter," 1049). In this case the plaintiff, an occupant of land adjacent to the defendant's premises, recovered judgment for injury to the land through the fact that smoke, soot, ashes, etc., were cast upon it in the defendant's process of cleaning its locomotive boilers by means of compressed air. A consulting engineer, who testified in the plaintiff's behalf, admitted that he had had no experience with locomotive boilers, but that they presented no problems with respect to such processes that do not equally apply to other boilers, and that he had made a special study of power and combustion, and had had experience in doing away with the evils of smoke, etc. The Supreme Court holds that he sufficiently qualified himself as an expert to enable him to testify that in such cases as this one, deposits of soot on premises near those upon which boilers are cleaned can be avoided by using brushes instead of blowers and by washing the smoke to remove injurious impurities. The court finds that this method has been found to have been effectively used by railroads and in stationary plants for 70 years, and that the compressed-air method was adopted merely to save time and expense. And it is further found that the damage which the plaintiff sustained could have been avoided by the defendant's discharging the noxious fumes through a high chimney, and that it is no defense to such a suit that the compressed-air method is in common use.

## PERSONALS

Edward Wegmann and A. G. Hillberg have taken offices in the South Ferry Building, New York City, to engage as consulting hydraulic engineers on water-works, water-power developments, sewer systems, irrigation and drainage projects. Mr. Wegmann was for more than 30 years connected with the construction of the Croton Water-Works. He was the last chief engineer of the Aqueduct Commissioners and subsequently for four years consulting engineer of the Department of Water Supply, Gas and Electricity. He is well known through his books—"Design and Construction of Dams" and "The Water Supply of the City of New York." Mr. Hillberg has been connected with large hydro-electric developments, notably the Mississippi River Power Co.'s plant at Keokuk, Iowa. For the past two years he has been associate editor of "Engineering Record," in charge of hydraulics.

## ENGINEERING AFFAIRS

**The Illinois Section of the American Water Works Association,** succeeding the Illinois Water Supply Association, held its seventh annual meeting at the University of Illinois, Champaign-Urbana, Ill., Mar. 9 to 11. The program included the following papers: "The Design and Operation of Intermittently Operated Water Purification Plants," E. B. Black, consulting engineer, Kansas City, Mo.; "Wash Water Salvage at Champaign and Urbana," H. E. Babbitt, instructor in University of Illinois; "Relation Between Bacteriological Standards and Vital Statistics at Hannibal, Mo.," W. F. Monfort, consulting chemist, St. Louis, Mo.; "Loss of Head on Strainers of Water Filters," Langdon Pearse, division engineer, sanitary district of Chicago; "Experiences in Rebuilding and Reinforcing a Water Works System," O. T. Smith, superintendent and manager, Water Works Co., Freeport, Ill.; "The New Harrisburg (Ill.) Filter Plant," L. F. Payne and Glen W. Bass, Central Illinois Public Service Co.; "Soft Water" (Illustrated), Cass L. Kennicott, vice-president and general manager, The Kennicott Co., Chicago; "Water Works Improvements at Springfield, Ill." (Illustrated), W. J. Spaulding, commissioner of public property, Springfield, Ill.; "Coal Mining Operations" (Moving Pictures), R. Y. Williams, director Miners' and Mechanics' Institute, University of Illinois; "Coal Resources of the Danville Area," F. H. Kay, assistant state geologist, Illinois; "Investigation of Artesian Water Supplies in the Chicago Area" (Illustrated), Frank De Wolf, director Illinois State Geological Survey; "The New Filtration Plant at Decatur, Ill." (Illustrated), Harry Ruthrauff, commissioner of public property, Decatur, Ill.; "The New Filtration Plant at Quincy, Ill. (Illustrated), W. R. Gelston, superintendent Citizens Water Works Co., Quincy, Ill.; "Kinks in the Control of Hypochlorite at Denver," W. W. DeBerard, western editor, "Engineering Record," Chicago; "River Sand as a Filter Medium," L. A. Fritze, city chemist, Moline, Ill.; "Choice of Alloys in Connection with Water Works Equipment," Horace Carpenter, engineer, Sanitary District of Chicago; "The Practical Value of Publicity to the Water Works Man," S. C. Hadden, associate editor, "Engineering and Contracting," Chicago; "Treatment of Water for Locomotive Use," W. A. Pownall, water engineer, Wabash Railway, Decatur, Ill.; "The Possibilities of Improved Water from Deep Wells in Northern Illinois," C. B. Williams, hydraulic and sanitary engineer, Chicago; "Water Supply of Longview, Texas," Paul E. Green, civil and sanitary engineer, Chicago; "Arsenic in Filter Alum," Edward Barton and A. N. Bennett, state water survey, University of Illinois; "Some Features of the Ontario Statutes and their Administration Affecting Water Supply and Sewerage," F. A. Dallyn, provincial sanitary engineer, Toronto, Canada; "The State Public Utilities Commission of Illinois and Water Works," W. A. Shaw, member Public Utilities Commission, Springfield, Ill.; "State Regulation of Municipally Owned Plants," C. M. Larson, chief engineer, Railroad Commission of Wisconsin; "The Illinois Utilities Commission and the Water Works Companies," C. G. Bennett, mechanical engineer, Illinois Utilities Commission, Springfield; "The Application of the Theories of Regulation to the Management of Utilities," Douglas A. Graham, principal assistant engineer, Dabney H. Maury, Chicago; "Economic Waste Aspects of Water Works

Operation," Ralph E. Heilman, assistant professor of Economics, University of Illinois; "Ancient and Modern Accounting for Utilities," Edward A. Pratt, president Edward A. Pratt Audit Co., Peoria, Ill.; "Water Waste Prevention by Individual Meter vs. District Meters," R. O. Wynne-Roberts, consulting engineer, Regina, Sask., Canada. The exhibits of the Associates were shown in Engineering Hall.

## NEW PUBLICATIONS

COX'S COMMERCIAL CALCULATOR. By Edward L. Cox. Published by Funk & Wagnalls Co., New York. 7x11 in.; 203 pages; cloth. Price, $10.

A collection of tables, from which the product of any two numbers whose sum does not exceed 202,000 can be found. The use of the tables is briefly explained and illustrated. Two key numbers must be obtained from the numbers to be multiplied, by the use of a simple formula. Each key number forms an unusual index to the table numbers required, in that the digits must be taken in order from right to left to find the correct page, column, section and line. The product desired is the difference of these two table numbers. The tables themselves are conveniently arranged and the book is valuable for commercial and scientific purposes in which products up to ten billion, accurate to the smallest unit, are essential.

GEARING. By A. E. Ingham. Published by D. Van Nostrand Co., New York. 5½x8½ in.; 181 pages; cloth. Price, $2.50

To the outsider the acquiring of practical gearing knowledge has been somewhat of a task. In one book he may find a complicated discussion of gearing principles; another may give him the manufacturing methods, while he must consult a third to find practical methods of design. Mr. Ingham has endeavored in the one book to cover the whole broad field of gearing. Nearly half of the 181 pages relate to spur gears, but in this section is stated the theory of the involute and cycloidal curves, which is later applied to the bevel, worm, spiral and helical gears.

The reader is given methods of drawing gear-teeth curves, such as the Brown & Sharpe single curve and the Grant odontograph; is shown how to calculate speed ratios for simple and compound gearing; and is informed regarding the materials and proportions of gears for the efficient transmission of power. The diametral, circular and metric systems of gear pitches are described. Numerous working drawings, cross-sections as a rule, accompany the design directions, while the manufacture and application of the various gears are illustrated by photographs. Formulas are given so that the reader may understand and pass through all the steps necessary to obtain the complete dimensions, but the numerous curves and tables, demonstrated by numerical examples, afford many short cuts. In general, Mr. Ingham's work offers a simple and adequate presentation of the information required in the design and application of gearing.

## BOOKS RECEIVED

OXY-ACETYLENE WELDING AND CUTTING. By Calvin F. Swingle. Frederick J. Drake & Co., Chicago, Ill. Cloth; 190 pages, 4¼x6¾ in.; 76 illustrations. Price, $1.

HANDBOOK OF MACHINE SHOP MANAGEMENT. By John H. Van Deventer. McGraw-Hill Book Co., Inc., New York. Leather; 374 pages, 4x7 in.; 244 illustrations. Price, $2.50.

ELEMENTARY ELECTRICITY AND MAGNETISM. By W. S. Franklin and Barry MacNutt. The Macmillan Co., New York. Cloth; 174 pages, 4¾x7¾ in.; 152 illustrations. Price, $1.25.

ADVANCED THEORY OF ELECTRICITY AND MAGNETISM. By W. S. Franklin and Barry MacNutt. The Macmillan Co., New York. Cloth; 300 pages, 5¾x8¾ in.; 217 illustrations. Price, $2.

SANITARY REFRIGERATION AND ICE MAKING. By J. J. Cosgrove. Technical Book Publishing Co., Philadelphia, Penn. Cloth; 331 pages, 5½x8½ in.; 103 illustrations; tables. Price, $3.50.

PREVENTING LOSSES IN FACTORY POWER PLANTS. By David Moffat Myers. The Engineering Magazine, New York. Cloth; 560 pages, 5x7½ in.; 68 illustrations, including several plates. Price, $3.

**Antimold and Antibug Varnish for Books**—The following ingredients make a varnish that has been found very useful in protecting books from mold and from roaches: Bichloride of mercury, 2 parts; orange shellac, 20 parts; oil of turpentine, 250 parts, and the balance up to 1000 parts, 95 per cent. ethyl alcohol.—"The Canal Record," Dec. 16, 1914.

## TRADE CATALOGS

Elliott Co., 6910 Susquehanna St., Pittsburgh, Penn. Bulletin G. Steam traps. Illustrated, 12 pp., 6½x10 in.

The Tracy Engineering Co., San Francisco, Calif. Catalog No. 10. Steam purifier. Illustrated, 20 pp., 6x9 in.

The Blaisdell Machinery Co., Bradford, Penn. Loose Leaf Catalog. Air compressors, vacuum cleaners. Illustrated, 6x9 in.

Semet-Solvay Co., Syracuse, N. Y. Pamphlet. Solvay 75% Calcium Chloride for refrigeration. Illustrated, 16 pp., 4x9 in.

B. F. Sturtevant Co., Hyde Park, Mass. Bulletin No. 206. Generating sets with vertical engines. Illustrated, 20 pp., 6½x9 in.

Lunkenheimer Co., Cincinnati, Ohio. Booklet. "Ferrenewo" valves. Illustrated, 10 pp., 3½x6 in. Booklet. "Clip" valves. Illustrated, 10 pp., 3½x6 in.

The Bristol Co., Waterbury, Conn. Bulletin No. 192. Bristol's long-distance electric transmitting and recording system. Illustrated, 8 pp., 8x10½ in.

Chicago Pneumatic Tool Co., Chicago, Ill. Bulletin No. E-35. Universal electric drills. Illustrated, 8 pp., 6x9 in. Form No. 212. Boyer riveting hammer. Illustrated.

Harrison Safety Boiler Works, Philadelphia, Penn. Cochrane Engineering Leaflet No. 17. "Reducing Boiler-Room Costs by Heating and Softening the Feed Water." Illustrated, 20 pp., 6x9 in.

Buffalo Forge Co., Buffalo, N. Y. Catalog No. 200. Planoidal Fans. Illustrated, 48 pp., 6x9 in. Catalog No. 201. Niagara conoidal fans. Illustrated, 64 pp., 6x9 in. Bulletin No. 182-E. Electric fans for blowing, ventilating, cooling, drying. Illustrated, 32 pp., 6x9 in.

Charles T. Main, engineer, Boston, Mass., is distributing a handsomely printed volume of halftone illustrations of industrial plants of his design. This follows a similar work issued some time since, and is inscribed "Industrial Plants, Vol. 2." Prospective builders of anything from cotton mills to central stations will find in it worth-while suggestions.

## BUSINESS ITEMS

The New River Co., equipment sales department, Macdonald, West Virginia, is sending out a list of second-hand power plant equipment, quoting prices and terms.

Harry J. Ernst, advertising manager of the D. T. Williams Valve Co., Cincinnati, Ohio, has been elected treasurer of the company, succeeding R. E. Mullane, recently elected president.

The Link Belt Co., Chicago, Ill., is sending out bulletins descriptive of the Wendell centrifugal coal drier, and the Link-Belt electric hoist. Both bulletins show illustrations of the equipment and go into details of construction. Copies are sent on request.

The Chicago office of the Terry Steam Turbine Co., Hartford, Conn., is now in charge of A. W. de Revere, located in the Peoples Gas Building. This company has also opened an office in the Michigan Trust Bldg., Grand Rapids, Mich., in charge of A. L. Searles.

The Sprague Electric Works of General Electric Co. has recently opened a branch sales office in Cleveland, Ohio. It will be in charge of Frank H. Hill, manager, who also has charge of the Pittsburgh office. The Cleveland office will be located in the Illuminating Building.

The sales of Sarco steam traps, Sarco vacuum valves, and Sarco temperature regulators have shown such substantial continued increases during the past half year, that the Sarco Engineering Co. of New York has moved its offices into larger quarters in the new South Ferry Building, 1 State St.

A very interesting booklet has just been published by the Murray Specialty Mfg. Co., 55 West Woodbridge St., Detroit, on "Boiler Feed—How To Regulate It." It's a 32 page booklet describing the apparatus in detail and containing many clear and instructive illustrations. Testimonial letters and installation details are also given. A request brings a copy.

The Harvard Medical School has recently given an order to the Builders Iron Foundry, Providence, R. I., for two extra heavy meter tubes for boiler feed service. This institution already has two Venturi meters on its heater conservation system and six Venturi meters for brine measurement. M. L. Bayard & Co., of Philadelphia, have recently placed an order for 36 double dial indicating instruments to be used in connection with effluent controllers at the Cleveland filtration plant.

The Southwark Foundry & Machine Co., Philadelphia, Penn., has secured the exclusive United States license to manufacture the Harris valveless engine, Diesel principle, which will hereafter be known as the Southwark-Harris valveless engine. The engine will be built in sizes from 75 b.hp. to 1000 b.hp. Leonard B. Harris, the inventor of the Harris valveless engine will be with the company as consulting engineer and naval architect, and J. P. Johnston will be in charge of oil engine sales.